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CHRYSLER CORPORATION

FINAL ENGINEERING REPORT
ON THE
LVTPX12
CONTRACT NObs 4777

VOLUME I
TECHNICAL STUDY

CHRYSLER CORPORATION
DEFENSE OPERATIONS DIVISION
P.O. BOX 757, DETROIT 31, MICHIGAN

CHRYSLER CORPORATION

FINAL ENGINEERING REPORT

ON THE

LVTPX12

CONTRACT NObs 4777

VOLUME 1

TECHNICAL STUDY

PREPARED FOR:
BUREAU OF SHIPS
DEPARTMENT OF THE NAVY
WASHINGTON, D. C. 20360

SUBMITTED BY:
CHRYSLER CORPORATION
DEFENSE OPERATIONS DIVISION
P. O. BOX 757
DETROIT 31, MICHIGAN

CHRYSLER CORPORATION

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SECTION 1.0
INTRODUCTION

INTRODUCTION

Chrysler Corporation is pleased to submit this final engineering report to the Bureau of Snips in accordance with the requirements of Contract NObs 4777.

This report is in three major divisions covering the following:

(1) A technical study report of the work done in Phase I of the LVTPX12 program.

(2) A management plan for the work to be conducted in Phase II and subsequent phases of the LVTPX12 program.

(3) A statement of cost information with regard to Phase II and subsequent phases of the LVTPX12 program.

The Phase I work described in the Technical Study portion of this report was completed in the eleven month period from June 1964 through May 1965. The engineering study covered preliminary design and model testing of a new personnel and cargo carrying tracked amphibian intended as the replacement for the LVTP5 family of vehicles in the 1970 decade.

The engineering study program was conducted along two basic design approaches to achieve the specified water speed, one approach utilizing the tracks alone for water propulsion, and the second approach investigating means other than the tracks for water propulsion. Accordingly, the two designs indicated in the frontispieces have been developed as optimum technical solutions to the problem statement contained in the Bureau of Ships Preliminary Specification for the LVTPX12.

Three problem areas were found to be of such significance as to merit special mention — water speed, armor protection, and the transmission.

It was recognized at the outset of the program that accomplishment of the water speed goal would be no easy task. The high priority assigned this performance characteristic was fully recognized accordingly, a model test program was completed in the Chrysler study which, to the best of our knowledge, is the most thorough and exhaustive ever undertaken in support of an amphibian design program. That model test program has yielded results which convince us that the track propelled vehicle concept we offer will, without question, meet the requirement for 8 MPH water speed. The second water propulsion approach employs twin screw propellers. Our model test work gives firm indication that this configuration will achieve a water speed of 10.7 MPH.

In the field of armor protection, recent developments in high hardness steel materials have permitted the required level of ballistic protection in a steel hull at a significant weight saving over an aluminum structure. Chrysler Defense Engineering has participated in steel armor developments in the past year, the product of which has been applied to the LVTPX12.

The search for a suitable transmission has produced convincing evidence that the design of a new transmission for the LVTPX12 should be undertaken by the Phase II contractor. This engineering report contains the preliminary design of the transmission recommended by Chrysler, and we would propose to develop the unit as a completely Government owned design.

Either of the two vehicle design approaches outlined in this report represent a substantial advancement in vehicle performance and system effectiveness when

compared to the LVTP5A1. Our technical design group has worked closely with procurement and manufacturing personnel to the end that the new amphibian can be produced at an economical cost and we assert that the LVTPX12 concepts presented are feasible vis-à-vis design, manufacturer and operational criteria.

Chrysler is capable and ready to continue participation in this important new vehicle program.

SECTION 2.0
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2.0 INDEX OF REQUIREMENTS

This final report on the work accomplished by Chrysler Corporation under Contract NObs 4777 is contained in seven individually bound books designated as follows:

- Technical Study: Volume I and Volume 2

  A complete expository report covering the engineering aspects of the concepts, their systems and components.

- Program Plan: Volume 3

  An analysis and recommendation for the organization, management, planning and scheduling of Phase II, and all succeeding program activities for the life of the vehicle program.

- Cost Estimates: Volume 4

  Cost Breakdown analyses for projected Phase II operations as well as budgetary estimates for all other related activities throughout the life of the program are presented.

- Appendices: Book I - Engineering data supporting the water performance report included in Volume I of the Technical Study.

  Book 2 - All other non-classified supporting information for the remainder of the report.

  Classified Book - Material referring to armor, night vision, and effectiveness analyses which require security classification. (This is the only one of the seven volumes which is security classified.)

Subjects treated in each of the volumes are segregated into appropriate sections as shown in the table of contents included in the front of each volume. Appropriate cross-reference notations are included in the text to facilitate location of supporting data. Charts, graphs, tables, drawings, illustrations and photographs are interspersed throughout the report and its appendices to support the material under discussion.

An understanding of the information contained in this report can be aided by the manner in which the information is presented. Furthermore, the format itself can be a tool for the wise evaluation of this information by the client. The contractor recognizes, however, that regardless of format, the results of this study will not be simple to evaluate. Evaluation will be made over a period of time and by several reviewers. To provide for such intermittent perusal, and to aid reviewers weighing or re-evaluating specific items, the following index is provided. This index is designed to show quickly the volumes and sections that include the information being sought.

Subjects or areas of interest are listed on the following pages as they appear in:

2.1 Work statement for Phase I, dated 29 January 1964.

2.2 Bureau of Ships Specification for LVTPX12, dated 31 January 1964.

In the following Index the locations of the information being sought is specified by volume, by section, and by the paragraph wherever possible. Some of the index subjects will be of the type that are covered throughout the report, a volume, or a section. In these cases an asterisk (*) will be

used to indicate that the subject can not be located in a specific paragraph but is covered in a general manner in the section, the volume, or throughout the report. If the subject being sought also appears in an appendix, the appendix designator and section are also indicated.

2.1 Work Statement for Phase I
(Dated 29 January 1964)

ITEM	SUBJECT	VOL	SECT	PARA	APPENDIX #	APPENDIX SECTION
1	Conduct a basic Engineering Study	*	*	*	*	*
1a	Selection of Standard Components	1 2	* *	* *	* *	* *
1b	Preliminary design of new items	1 2	* *	* *	* *	* *
1c	Preliminary installation and arrangements	1 2	* *	* *		
1d	Preliminary design of hull form	1 1	4.0 7.0	4.2 4.5 4.10 7.1	A	4.0
1e	Scale model test	1	4.0	4.2 4.3 4.4	A	6.0 14.0
2	Prepare final engineering report	*	*	*	*	*
2a	Inboard and outboard profiles	1 1	5.0 4.0	5.5 4.2 4.5		
2b	General arrangement drawings	1	5.0	*		
2c	Machinery arrangement	1 2	* *	* *		
	. Power plant and power train	1 1 2	8.0 5.0 9.0	* * 9.[illegible]		
	. Inlet air	2	14.0	14.4.1	H	*
	. Exhaust system	2	14.0	14.4.2	H	*
	. Cooling System	2	15.0	15.3	I	*
	. Land-Water locomotion system	1 1 2	8.0 4.0 9.0	* 4.2 4.6 *	 A	 6.0 15.0
2d	Hydrodynamic drawings	1	4.0	4.2 4.5		

2.1 Work Statement for Phase I (Continued)

ITEM	SUBJECT	VOL	SECT	PARA	APPENDIX	
					#	SECTION
2e	Hydrostatic and stability curves	I	4.0	4.8	A	*
2f	Predicted land characteristics	I	8.0	8.10	D	7.3
		2	9.0	9.10		
2g	Predicted water characteristics	I	8.0	8.10	D	4.0
		I	4.0	4.2	A	*
		I	4.0	4.5	A	*
		I	4.0	4.10	A	*
2h	Fuel consumption	2	13.0	13.1	G	2.0
					D	5.0
					D	4.0
	. Land endurance	2	13.0	13.1	G	2.0
					D	5.0
	. Water endurance	I	4.0	4.10	D	5.0
		2	13.0	13.1	G	2.0
2i	Machinery list	3	4.0	*	Q	*
2j	Weight estimates	I	*	*		
		2	*	*		
		2	21.0	21.2	N	*
2k	Method of operation	I	4.0	4.6	A	15.0
		I	5.0	*		
		2	20.0	20.2		
	. of control	I	4.0	4.6	A	15.0
		I	5.0	*		
		I	8.0	8.8		
				8.9		
		2	20.0	20.2		
2l	Data and results of model test	I	4.0	4.2-4.7	A	14.0
2m	PERT-COST system	3	3.0	*		
		4	*	*		
	. Work breakdown structure	3	3.0	*		
			10.0			
		4	2.0	*		
	. Contract cost estimates	4	*	*		
2n	Estimated cost breakdown	4	*	*		

2.1 Work Statement for Phase I (Continued)

ITEM	SUBJECT	VOL	SECT	PARA	APPENDIX #	APPENDIX SECTION
2n(1)	. 25 pilot production vehicles	4	4.0	*		
2n(2)	. Special tooling	4	5.0	*		
	. Pre-production	4	5.0	*		
	. Production	4	5.0	*		
2n(3)	. 1000 production vehicles	4	6.0	*		
2n(4)	. Vehicle design agent estimate	4	3.0	*		
2n(5)	Facilities required	4	*	*		
2o	Time-cost-performance trade-offs	1	*	*		
		2	*	*		
2p	Recommendations for changes	1	*	*		
		2	*	*		
		3	*	*		
		4	*	*		
2q	Comments on schedule	3	10.0	*		
		4	*	*		
2r	Foreseeable technical problems	1	*	*		
		2	*	*		
2s	Proposed SP/M program	3	6.0	*		
2t	Contractor management commitments	3	*	*		
2t(1)	. Program management structure	3	2.0	*		
2t(2)	. Program management personnel	3	2.0	*		
2t(3)	. Management control techniques	3	3.0	*		
		3	4.0	*		
2t(4)	.. Planned subcontracts over $10,000	3	4.0			
		4	2.0	2.1.4		
2t(5)	. List of facilities required	4	2.0	2.1.3		
2t(6)	. Capabilities for VDA	3	7.0	7.10		

2.1 Work Statement for Phase 1 (Continued)

ITEM	SUBJECT	VOL	SECT	PARA	APPENDIX #	APPENDIX SECTION
2u	Contract recommendations	3	3.0	3.4		
3	Monthly progress reports	*	*	*		
4	Two review meetings	*	*	*		
5	Full scale mock-ups	1	3.0	*		
6	Proposed prototype test agenda	3	5.0	*	0	*
7	Proposed instrumentation drawings	3	5.0	*		

2.2 Bureau of Ships Preliminary Specification
Assault Amphibian Personnel Carrier (LVTPX 12)
(31 January 1964)

ITEM	SUBJECT	VOL	SECT	PARA	APPENDIX #	APPENDIX SECTION
3	Requirements	1 2 3 4	* * * *	* * * *		
3.1	Materials	1 2	* *	* *		
3.1.1	Commercial items	1 2	* *	* *		
3.1.2	Standard military components	1 2	* *	* *		
3.1.3	Parts marking	*	*	*		
3.2	Interchangeability	1 2	* *	* *		
3.3	Ease of maintenance	1 2 1 2 2 2	* * 4 9.0 16.0 20.0	* * 4.6 9.11 16.5 20.7	 A	 15.0
3.4	Lifting devices	1 2 1 2	7.0 20.0 * *	7.4 20.6 * *	C	1.0
3.5	Removable components	1 2	* *	* *		
3.6	Lubrication	1 2 1 1 1 1	* * 8.0 8.0 9.0 20.0	* * 8.8.6 8.9.6 9.9.6 20.7		
3.7	Fittings	1 2	* *	* *		

2.2 Bureau of Ships Preliminary Specification (Continued)

ITEM	SUBJECT	VOL	SECT	PARA	APPENDIX #	APPENDIX SECTION
3.8	Maintenance tools	1	*	*		
		2	*	*		
		3	6.0	6.2		
		3	7.0	7.2.8		
3.9	Parts and assemblies	1	*	*		
		2	*	*		
3.10	Selection of common hardware	1	*	*		
		2	*	*		
3.11	Performance characteristics	1	3.0			
		1	5.0	5.12		
		1	*	*		
		2	*	*		
3.11.1	Speed					
	a) Land-forward & reverse	1	8.0	8.10	D	7.3
		2	9.0	9.1		
				9.2		
	b) Water speed	1	8.0	8.10		
		1	4.0	4.2	A	1.0 thru 15.0
		1	4.0	4.5		
		1	4.0	4.10		
		2	9.0	9.7	A	24.0
3.11.2	Braking ability	1	8.0	8.7	D	4.3
						4.4
3.11.3	Climbing ability	1	8.0	8.10		
		2	9.0	9.11		
3.11.3.1	Forward slope	1	*	*		
		2	*	*		
		1	8.0	*		
		2	9.0	9.10		

2.2 Bureau of Ships Preliminary Specification (Continued)

ITEM	SUBJECT	VOL	SECT	PARA	APPENDIX #	APPENDIX SECTION
3.11.3.2	Side slope	1 2 1 2	* * 8.0 9.0	* * * 9.10		
3.11.4	Maneuverability	1 1 1 2	4.0 4.0 5.0 9.0	4.6 4.10 * 9.10	A	4.0
3.11.4.1	OFF-Road operation	2	9.0	9.10		
3.11.4.2	ON and OFF Ship Operation	2 1	20.0 4.0	20.2 4.10		
3.11.4.3	Alongside ship operation	2 1 2	20.0 4.0 9.0	 4.10 9.7		
3.11.5	Stability	1 2	4.0 9.0	* 9.10	A	16.0-20.0
3.11.5.1	Water	1	4.0	4.8 4.10	A	16.0-18.0
3.11.5.2	Land	2	9.0	9.10		
3.11.6	Trench-Crossing ability	2	9.0	9.10		
3.11.7	Vertical-obstacle ability	2	9.0	9.10		
3.11.8	Surfing ability	1 1 2	4.0 * *	4.10 * *		
3.11.9	Hoisting characteristics	1 2 1	* * 5.0	* * 5.12 5.14	 N	 5.0
3.11.10	Endurance (fuel usage)	1 2	8.0 13.0		D G	4.0 *

2.2 Bureau of Ships Preliminary Specification (Continued)

ITEM	SUBJECT	VOL	SECT	PARA	APPENDIX #	APPENDIX SECTION
3.11.11	Climatic conditions	1 2	* *	* *		
	a) Below -25°	2 2	17.0 9.0	17.3 9.1	J	3.0
3.11.12	Service products to be used	1 2	* *	* *		
3.11.13	Fuel fill rate	2	13.0	13.2.5		
3.11.14	Operating life	1 2	* *	* *		
3.11.15	Winterization kit	2	17.0	17.3	J	3.0
3.11.16	Storage					
3.11.17	Electronic interference suppression	2	11.0	*		
3.11.18	Refueling at sea	2	13.0	13.2.5		
3.11.19	Auxiliary starting	2	17.0	17.12		
3.11.20	Performance curves	1	8.0	8.10	D	7.3
3.11.21	Vehicle detection	1 2	* *	* *		
3.12	Physical characteristics	1 2	* *	* *		
3.12.1	Hull	1	7.0	7.4	C	*
	a) Length	1 1 1	4.0 5.0 7.0	4.2 5.14 7.4	A N	
	b) Beam	1 1 1	4.0 5.0 7.0	4.2 5.14 7.4	A N	
	c) Height	1 1 1	4.0 5.0 7.0	4.4 4.10 5.14 7.4	A N	

2.2 Bureau of Ships Preliminary Specification (Continued)

ITEM	SUBJECT	VOL	SECT	PARA	APPENDIX	
					#	SECTION
	d) Capacity (personnel)	1 2	5.0 20.0	5.13 20.4	N	
	e) Capacity (cargo)	1 2	5.0 20.0	5.12 20.5	N	
	f) Compartment dimensions	1 1 2	5.0 7.0 20.0	5.12 7.4 20.5	N	
	g) Firing ports	1 2	5.0 20.0	5.13 20.1	N	
	h) Road clearance	1 2	5.0 9.0	5.13 9.9	N	
	i) Angle of approach	1 2	5.0 9.0	5.13 9.10	N	
	j) Angle of departure	1 1 1 1 2	4.0 4.0 4.0 5.0 9.0	4.2 4.5 4.10 5.12 9.10		
	k) Safety rails and grab handles	1 2	7.0 20.0	7.4 20.3		
	l) Crew access hatches	1 1 2	5.0 7.0 20.0	5.13 7.4 20.1		
	m) Litter kit	2 2	17.0 20.0	17.4 20.5		
	n) Bilge drain plugs	1 2	7.0 10.0	7.4 10.2	 E	 *
3.12.1.1	Reinforcements	1	7.0	7.4	C	*
3.12.1.2	Fabrication	1	7.0	*		

2.2 Bure[illegible] of Ships Preliminary Specification (Continued)

ITEM	SUBJECT	VOL	SECT	PARA	APPENDIX #	APPENDIX SECTION
3.12.1.3	Armor, armament and ABC protection	1	5.0	*		
		1	6.0	*		
		2	16.0	*		
		2	17.0	17.14		
	a) Armor protection	1	6.0	*	B	
		1	7.0	7.3.2		
		1	5.0	5.13		
	b) Armament	2	16.0	*		
		1	5.0	5.13		
	c) Chemical protection	2	17.0	17.9		
	d) Nuclear explosion protection	2	19.0	*		
	e) Vision devices	1	*	*		
		2	*	*		
		2	17.0	17.8	K	2.0
		2	18.0		L	*
3.12.1.4	Ramp	1	4.0	4.2		
				4.5		
			5.0	5.6		
			10.0	10.3		
			12.0	12.4		
			7.0	7.4	C	*
3.12.1.5	Troop seats	1	5.0	5.13		
		2	14.0	*		
		2	17.0	17.5		
		[illegible]	20.0	20.4		
3.12.1.6	Miscellaneous provisions					
	.1 - towing devices	1	7.0	7.4	C	1.0
		2	20.0	*		
	a) Quick release tow hitch	1	7.0	7.4		
		2	20.0	20.6	C	1.0
	b) Towing pintle	1	7.0	[illegible]	C	1.0
		2	20.0	20.[illegible]		
	c) Towing cable	2	20.0	20.6	C	1.0

2.2 Bureau of Ships Preliminary Specification (Continued)

ITEM	SUBJECT	VOL	SECT	PARA	APPENDIX #	APPENDIX SECTION
	d) Pad eyes	1 2	7.0 20.0	7.4 20.6	C	1.0
	.2 - Tie down	1	7.0	7.4		
	.3 - Lifting devices	1 2	7.0 20.0	7.4 20.6	C	*
	.4 - Mooring devices	1 2	7.0 20.0	7.4 20.6	C	*
	.5 - Cargo handling system	1 2	7.0 20.0	7.4.3 20.5		
3.12.1.7	Boarding steps	1 2	7.0 20.0	7.4 20.1		
3.12.1.8	Identification plate	2	20.0	20.1		
3.12.1.9	Land-Locomotion system	2	9.0	9.7		
3.12.2	Machinery	1 2	* *	* *		
3.12.2.1	Propulsion system	1	8.0	*	O	*
3.12.2.2	Power train	1	8.0	*	D	*
3.12.2.3	Fuel system	2	13.0	*	G	*
3.12.2.4	Lubrication system	1 1	8.0 8.0	8.8.6 8.9.6		
3.12.2.5	Bilge system	2 2 2 2	10.0 11.0 12.0 20.0	10.2 11.4.6 12.4.2 20.3	E F	* 1.0
3.12.2.6	Hydraulic and lubrication lines	2	12.0	*		
3.12.2.7	Controls					
	.1 - Engine throttle control	1 2 2	8.0 18.0 20.0	8.8.1 8.9.1 18.8.3 20.2		

2.2 Bureau of Ships Preliminary Specification (Continued)

ITEM	SUBJECT	VOL	SECT	PARA	APPENDIX	
					#	SECTION
	.2 - Steering control	1	8.0	8.8.2 8.9.2		
		2	18.0	18.8.3		
		2	20.0	20.2		
	.3 - Brake control	1	8.0	8.8.2 8.9.2		
		2	18.0	18.8.3		
		2	20.0	20.2		
3.12.2.8	Instrument panel	2 2 2	11.0 18.0 20.0	11.4.4 18.8.2 20.1		
3.12.2.9	Fire extinguishing system	2 2	17.0 20.0	17.1 20.3	J	2.0
3.13	Electrical	2	11.0	*		
3.14	Signalling Searchlight	2	17.0	17.6		
3.15	Inside lights	2 2	11.0 20.0	11.4.4 20.4		
3.16	Escape hatches	1 1 2	5.0 7.0 20.0	5.13 7.4 20.1		
3.17	Stowage	2 2	17.0 20.0	17.5 20.5		
3.18	Night vision viewing equipment	2 2	17.0 20.0	17.8 20.4	L	*
3.19	Navigation and communication	2	17.0	17.7 17.11		
		2	20.0	20.7		
3.20	Painting	-	-	-		
3.20.1	Surface preparation	-	-	-		
3.20.2	Priming and painting	-	-	-		

CHRYSLER CORPORATION

2.2 Bureau of Ships Preliminary Specification (Continued)

ITEM	SUBJECT	VOL	SECT	PARA	APPENDIX	
					#	SECTION
3.21	Multi-source requirements	3	6.0	6.3		
3.22	Repair parts	4	2.0	*		
3.23	Technical documentation	3	*	*		
3.23.1	Sketches and layouts	3	4.0	*		
3.23.2	Drawings	3	4.0	*		
3.23.2.1	Class 1 drawings	3	4.0	*		
3.23.2.2	Class 2 drawings	3	4.0	*		
3.23.3	Associated list	3 3	4.0 4.0	* *		
3.24	Technical manuals	3	7.0	*		
4.0	Quality assurance provision	3 3	4.0 6.0	* *	 0	 *
5.0	Preparation for delivery	3	5.0	*		

SECTION 4.0
ANALYSIS OF WATER PERFORMANCE

CHRYSLER CORPORATION

4.0 ANALYSIS OF WATER PERFORMANCE

The LVTPX12 is to be an assault amphibious personnel and cargo carrier used in landing operations of the U. S. Marine Corps. It must be versatile and flexible, ready to embark on and to disembark from a landing ship, to move rapidly to shore, and there to operate as an armored land vehicle. Not withstanding the doctrine of eighty percent land operation and twenty percent water operation, water travel is a critical part of the operation and therefore a primary determinant of the success of the task.

Because of the importance of good water performance, special emphasis must be placed on this aspect of design. The subject of water performance includes all properties of the craft as a body floating in water or moving through it. Under this subject come:

- Speed
- Manueverability
- Sea keeping
- Surf ability
- Endurance
- Periodic motions
- Behavior on landing
- Vulnerability when lying alongside
- Range of stability
- Initial righting moment
- Moment to trim
- Pounds per inch immersion
- Particulars of form and corresponding curves

- Propulsive devices
- Required delivered horsepower

Some of these properties refer to the vehicle in a static condition, some are dynamic. Some can be calculated directly, some can be predicted by analysis, others can be determined only from model tests or from comparisons with other vehicles of similar characteristics. The difficulty of simulation and the lack of data from previous observations of tracked amphibians require some predictions to be approximate, periodic motions being an example.

In this section, the general principles of water performance, the specific problems relating to tracked amphibians, and the predictions for the LVTPX12 will be examined. The results of the model tests will be reviewed and compared in detail. The following topics will be considered in approximately this order:

- Desired Characteristics of the LVTPX12
- The Model Testing Program
- Model Tests of Auxiliary Propulsion
- Height of Bow Waves
- Conclusions From Tests
- Steering and Maneuvering
- Problems Meriting Further Research
- Static Particulars of the LVTPX12
- Dynamic Behavior
- Summary of Findings for Design of a Watergoing Vehicle.

All of these topics will be discussed as much as possible in simple physical terms. The mathematical analyses and derivations will be found in the Appendix

to this section (Appendix A).

Appendix A to this section on water performance is a necessary part of this test. In particular, a number of terms occurring in Section 4, such as "choker plates", will be confusing unless an understanding of such terms has previously been acquired. The definitions of common terms used in naval architecture have been included.

Appendix A also contains theoretical reviews, mathematical analyses, and all numerical calculations relating to the LVTPX12 as a floating body.

4.1 <u>Desired Characteristics of the LVTPX12</u>. The performance of the LVTPX12 as prescribed in Request Number 529-41431(s), insofar as speed and behavior in water are concerned, with a full load of fuel and complete OEM, crew of three with combat rations, arms, and water for 48 hours, plus full cargo of 10,000 pounds or a minimum of 25 fully equipped troops, shall be as follows:

- Desired speed ahead: 10 MPH
- Required speed ahead: 8 MPH
- Required speed backing: 3.5 MPH
- Maneuverability: "maximum attainable"
- Positive righting moment: to at least 90 degrees in light condition
- Endurance: 7 hours at 8 MPH
- Seaworthiness: safe for 10 foot plunging surf

- Launching: capable of embarking on or disembarking from flooded LSD or LPD either ahead or astern
- Lying alongside: safely maneuverable alongside ships or piers while handling cargo through deck hatches.

A summary of compliance with these specifications will be found in Paragraph 4.10. The summary shows that Chrysler Corporation has considered the stated specifications as minimum requirements rather than as limits of design, that in each case the specified water performance has been supplied, and that in many cases highly desirable characteristics not contemplated in the specifications have been achieved.

4.1.1 Implications in the Specified Characteristics. Although freeboard is not mentioned in the specifications, the maximum height of 8 feet 6 inches from the groundline coupled with the unavoidable displacement, sets a limit on maximum possible freeboard. The maximum safe speed of the vehicle is thus fixed for certain conditions of displacement and trim (See Paragraph 4.4). Likewise, the reserve buoyancy and range of stability are affected.

The required maximum length of 26 feet and the speed of 8 to 10 MPH result in a Taylor quotient from 1.37 at 8 MPH to 1.72 at 10 MPH (F_n from .41 to .517) depending on how much of the overall length can be designed into the waterline length.

If the craft is to lie alongside a ship or pier, vulnerable appendages or propulsive devices must be avoided. Yet the vehicle must be able to come alongside, stop, go forward, or back away. Likewise, the requirement of ability to negotiate high plunging surf demands invulnerability from the bars

or reefs that must necessarily exist if surf is to exist. Any appendages or propulsive devices must be operable from protected positions while the craft is in violent motion in shallow water. Again, the requirement that the vehicle be able to enter or leave its restricted space on a flooded LSD demands that some propulsive effort be provided even with the auxiliary propulsive devices fully housed.

The minimum allowable angles of approach and departure establish boundaries in the design of bow and stern for minimum resistance. This minimum angle of 35 degrees, plus the requirement that the vehicle be able to encounter and climb a vertical obstacle 36 inches high, prohibit the design of a bow with the large deadrise that would be sought for minimum resistance. It will not be possible to recess the front track idler entirely inside the hull. The tracks must protrude beyond the hull, thus presenting a discontinuity in the streamlined flow around the bow.

4.1.2 Desirable Characteristics Not Specified. The ideal vehicle should handle like a boat while operating in water. Nothing could be more desirable than sure and direct response to the helm, and such behavior would certainly produce increased re~' propulsive efficiency provided that the steering device did not sap a proh 'itive share of available power from the engine. Therefore, incorporating a means for reactive steering into the design is well worth study. (See Paragraph 4.6.2).

4.2 Model Testing Program. The object of the series of model tests was more than simply ɔ find a design that would go at maximum speed; it was also to observe general hydrodynamic principles in the resistance of heavy

tracked amphibians, to discover effects that increase speed, that decrease speed, or that produce little effect. As little time as possible was spent in testing principles adequately reported in the literature. Some ideas previously tested, however, have not been adequately reported. Some ideas, both fruitful and worthless, had never been tested before. Attention will be called to leads meriting further research in Paragraph 4.7.

The order of this review and analysis of the model tests will be:

- Description of the models.
- Meaning of propulsive efficiency and developed horsepower.
- The effects of model size and type.
- Experiments concerning track design.
- Effect of the return tracks.
- Stern planes, contravanes, and stern baffles.
- Bow planes, both submerged and above water.
- Form of ends of the vehicle.
- Effect of reduced beam.
- Side skirts.
- Effect of trim.
- Resistance and weight.
- Backing speed.
- Supplemental thrust and track propulsion.
- Slip
- Articulating, feathering grousers.
- Optimum tracked configurations.

The review of model tests of the propeller driven vehicle are in Paragraph 4.3. Certain technical terms and names of appendages are defined in Section 1.0, Appendix A. Photographs of the model with various bow and stern sections and with the appendages are provided with the text. The model tests are grouped in logical families to show the effects of modifications.

The tests were performed by the Ship Hydrodynamics Laboratory, University of Michigan. A detailed report on the facilities of the Laboratory may be found in Reference 22, Appendix A. The reports of the Laboratory are included in Appendix A, Section 14.0.

4.2.1 Description of the Models of the LVTPX12. Including tests on a wood model before the LVTPX12 proposal and the tests on a metal self-propelled model during this program, more than 100 tests were made. The wood model was 1/5 scale, while the scale of the aluminum self-propelled model was 1/4.5. The metal model was in three sections; the middle body, shown in Figure 4-1, exactly as long as allowable by the supports for the tracks; several bow sections; and several stern sections, as shown in Figures 4-2 through 4-9. The variation in length resulting from the use of these several combinations was insignificant, a maximum of 2.5 inches on the model. Because of the effect of changing the rake of bow and stern sections, it is not possible to keep the waterline lengths exactly the same for various trimming conditions, while keeping overall length constant.

The displacement most used for the tests was 45,000 pounds.

At this displacement, the prototype of the model would have had the following dimensions:

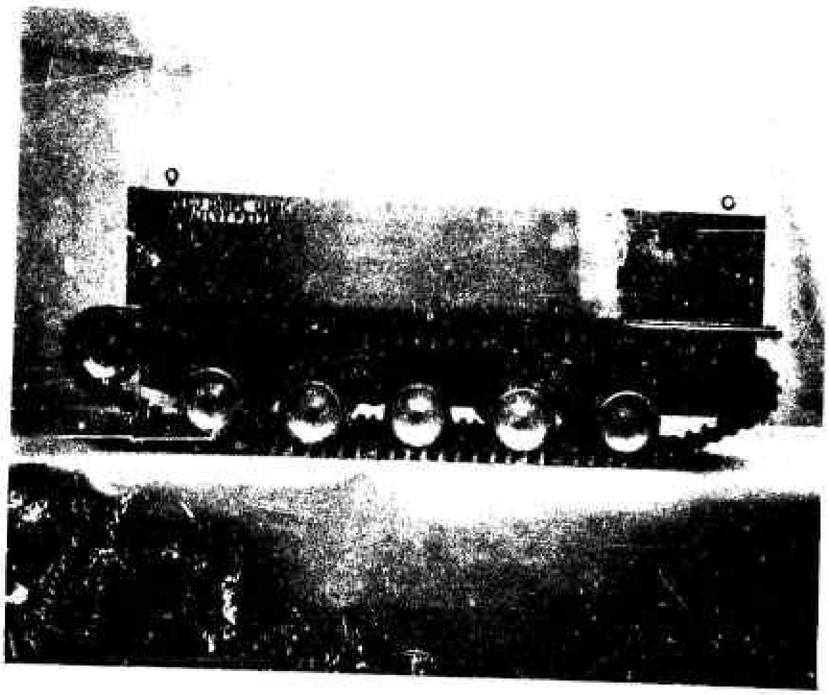

Figure 4-1 Middle Section of Model

Figure 4-2 Bow Section #1

Figure 4-3 Bow Section #2

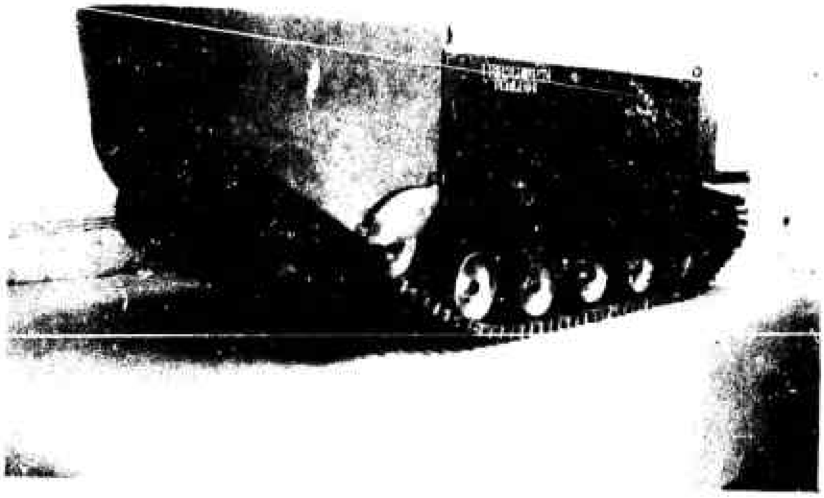

Figure 4-4 Bow Section #3

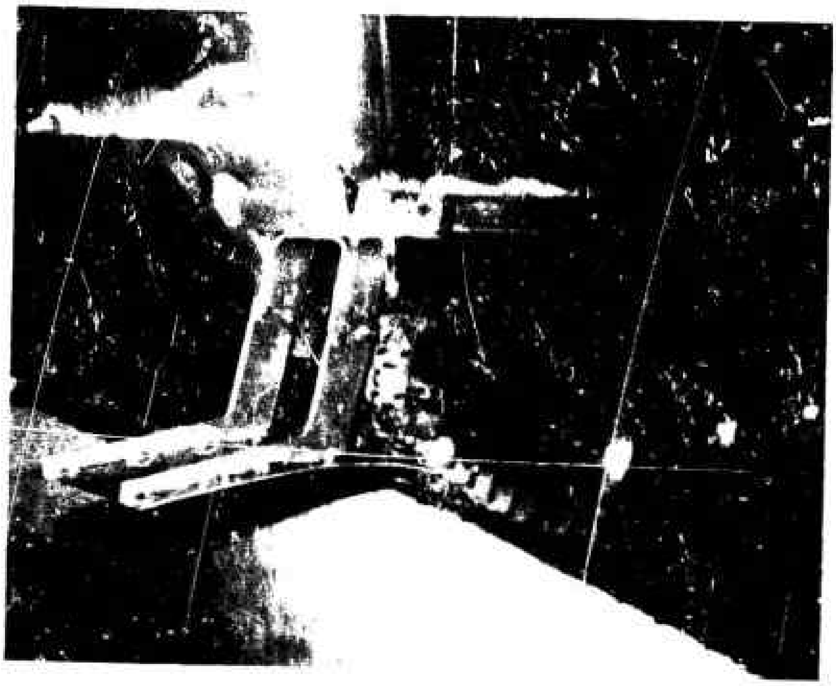

Figure 4-5 Stern Section #1

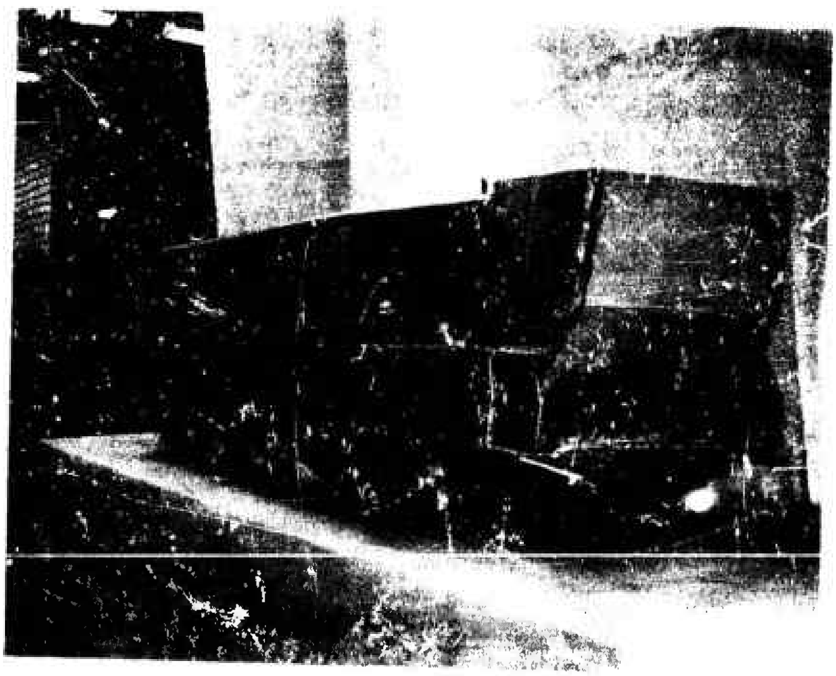

Figure 4-6 Stern Section #2

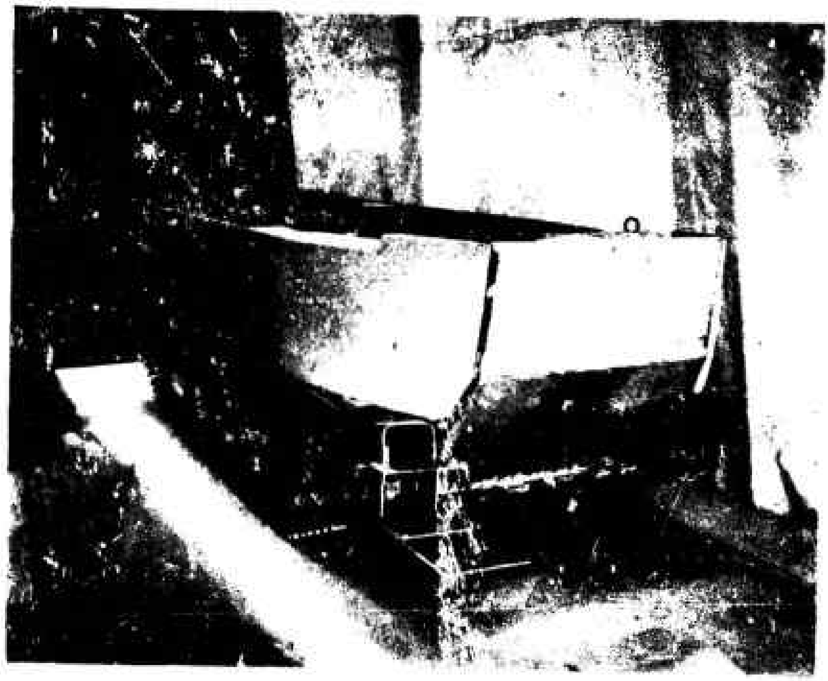

Figure 4-7 Stern Section #3

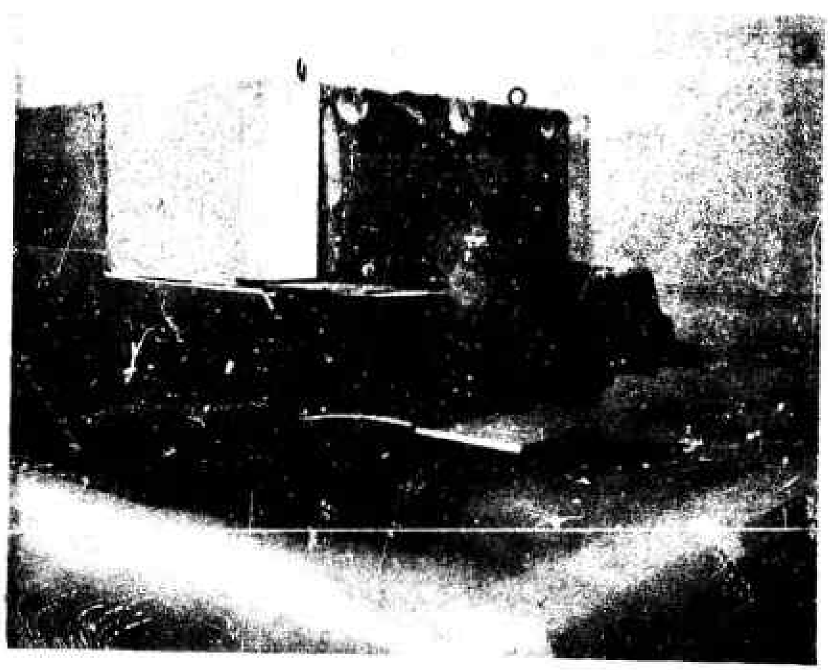

Figure 4-8 Stern Section #4

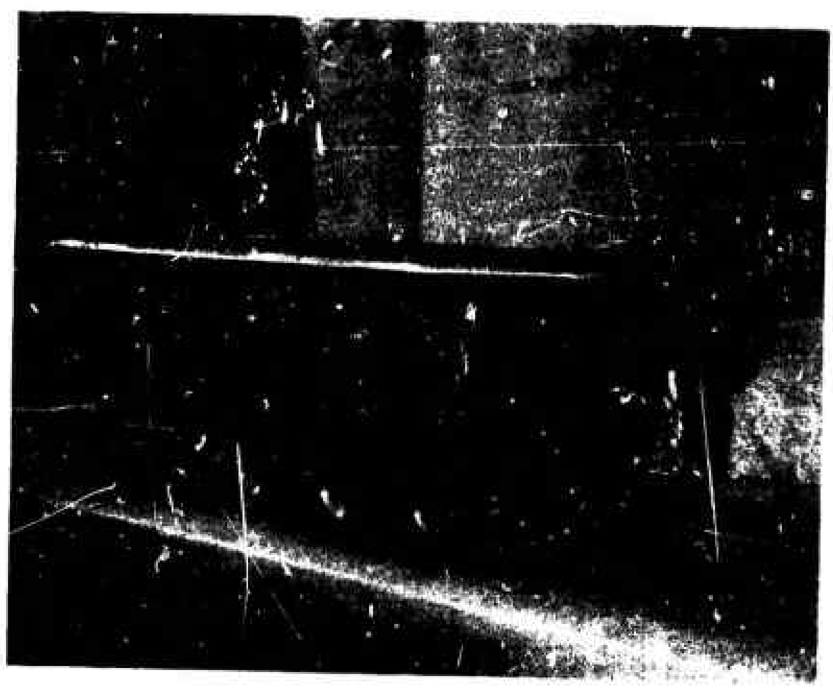

Figure 4-9 Stern Section #5

- Length, waterline, with Bow #1, 25.2 ft.
  with Bow #2, 24.65 ft.
  with Bow #3, 24.26 ft.

The above lengths are for zero trim, still water. The beam of the prototype was 10.3 feet.

For purposes of comparison, a standard length of 25 feet is applied. This results in no error in resistance of the prototype, because the length will correspond. The self-propelled model was driven by a DC motor with a Ward-Leonard speed control through a rheostat operated on the carriage. The motor drove a worm gear through a Lebow torque pick-up, so that the torque measured was directly on the motor output shaft. Periodically during the test, the model would be hoisted in air and a frictional torque line prepared for speeds from 500 to 3,000 RPM. The frictional torque was then subtracted from observed torque, yielding the net torque for net delivered horsepower.

The aluminum model was modified (Compare Figure 4-9 with 4-10 and 4-11) to provide the side wells for the propellers. The same power system was used in the propeller tests as for the tracked model. The side propellers were driven through flexible shafts which trailed in the water ahead of the screws, but the small resulting augment of resistance has been accounted for in the analysis.

After the wood model was used in tests of towing resistance and self-propelled tests with a Kort nozzle, (Figures 4-12 and 4-13) its beam was reduced by 20 percent to learn the effect of reduced beam on resistance. (See Figures 4-14 and 4-15).

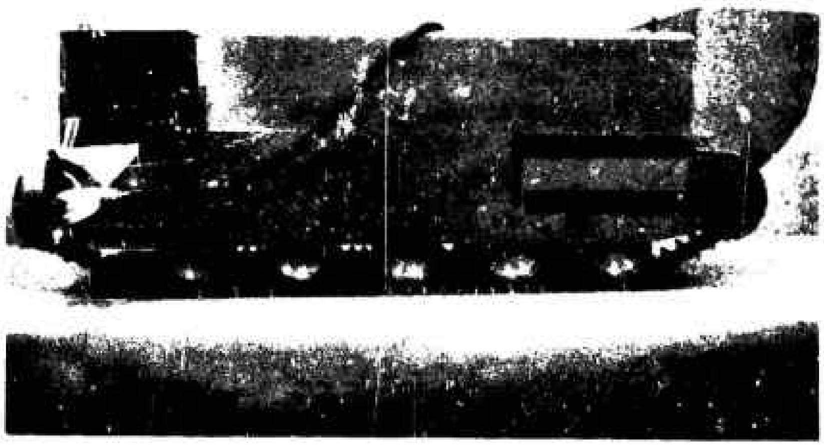

Figure 4-10 Model with Side Propellers

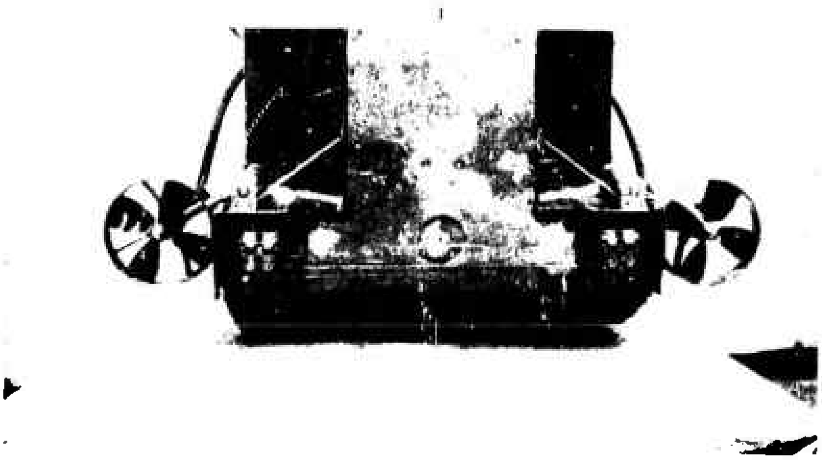

Figure 4-11 Stern View of Twin Propellers

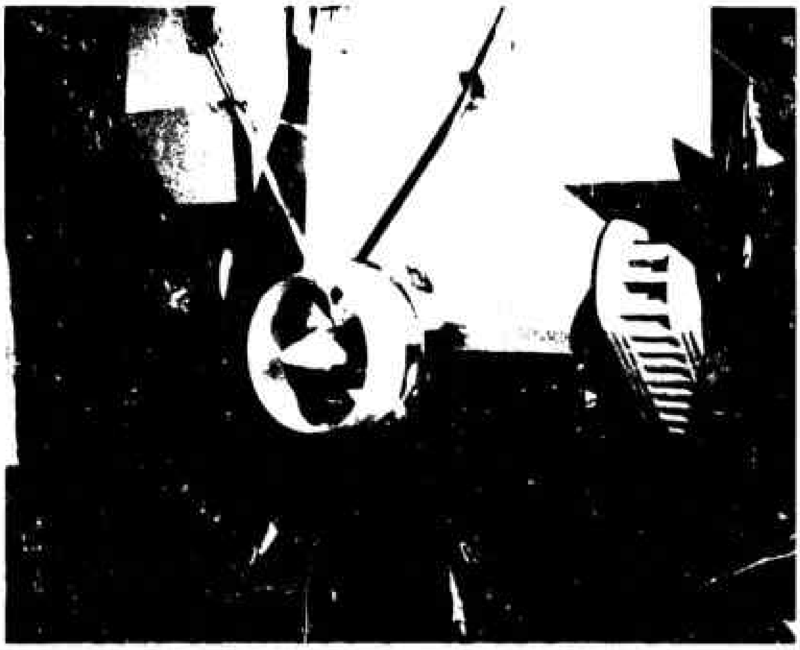

Figure 4-12 Single Kort Nozzle Propeller

Figure 4-13 Depth of Single Propeller Below Bottom

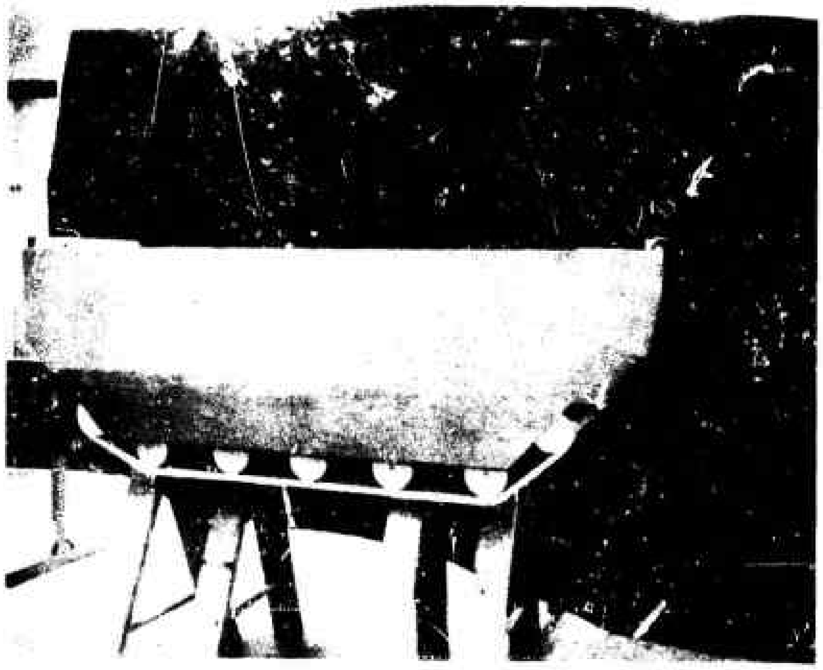

Figure 4-14 Side View of 1/5 Scale Wood Model

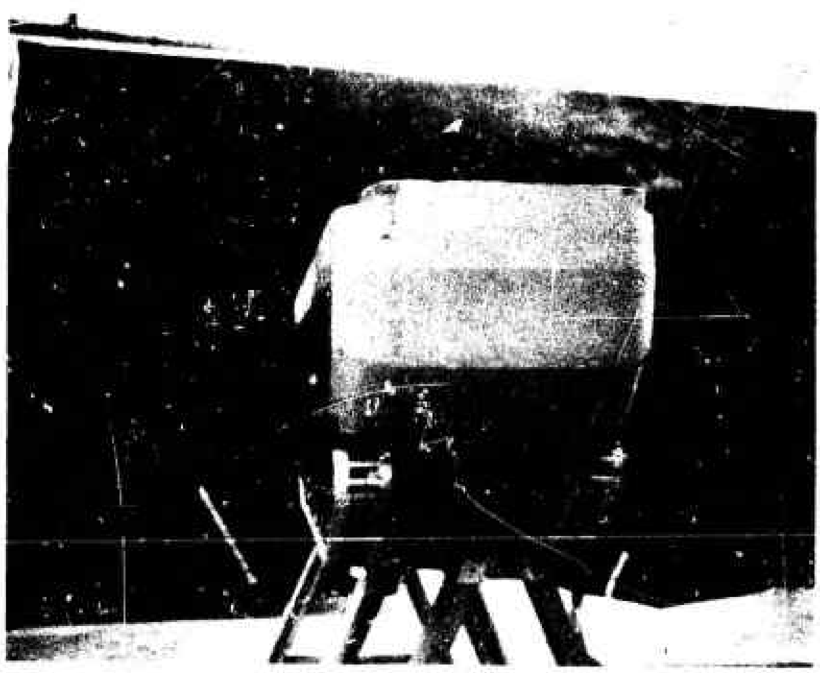

Figure 4-15 Bow View of Wood Model After Reducing Beam by 20%

4.2.2 Propulsive Efficiency and DHP. In the testing of ships' models, common practice is to obtain the resistance of the model with the propeller(s) removed, and from this resistance to calculate effective horsepower, EHP. The power required to turn the propulsive device, after subtracing all torque from mechanical friction, yields delivered horsepower, DHP. The propulsive coefficient is then the ratio, EHP/DHP. Tracked amphibians, however, lose all their characteristics when the tracks and suspension system are removed. In the naval architecture of these craft, the practice of obtaining the EHP with the tracks on and not turning has become standard by tacit consent. If it seems that one way to eliminate the high resistance of the suspension system would be to obtain the EHP with the tracks turning at vehicle speed (zero slip), a little reflection will show that this method would not provide any better insight into the behavior of the propulsive device, but would add to the complexity of tests and to the opportunity for error.

Therefore, in the report, the meaning of the propulsive coefficient will be the EHP with tracks locked, divided by DHP.

4.2.3 Importance of Model Size and Type. The very first principle of model testing is geometric similitude. This seems to be an elementary statement, yet in the past much reliance has been placed on models of tracked amphibians that were only approximately similar to their prototypes. They corresponded in gross shape to the prototypes, but details were only symbolized if duplicated at all. On one model of the LVTPX7, the tracks were represented by smooth boards (Reference 14). Beal reported that the resistance of a block model of the LVTPX2 was only 84 percent of the resistance of a more exact tracked model.

A wooden model of the LVTPX11 on a 1/5 scale showed an effective horsepower of 37.5 at 6.5 MPH, while the EHP for a 1/4 scale metal model with real tracks and grousers was 42 (Compare Re...ences 21 and 22.) At 8 MPH, a 1/16 scale wooden model of the LVTPX11 showed 65 EHP, while a 1/5 scale wooden model showed 75 EHP. Figure 4-16 is a comparison of the towing resistance between a 1/5 scale wooden model of the LVTPX12 and the metal, self-propelled model with real tracks. The metal model was fitted with a slightly more rounded bow than the wood model and therefore should have had slightly less resistance, not more.

More examples of such results can be found in the literature. The evidence is clear that misleading data will be obtained with extremely small models and with models not exactly similar in detail to their prototypes. This duplication of details applies particularly to the undercarriage. Hence in model construction:

- The model must be as large as the towing tank will accommodate, on a scale 1/5 or larger.
- Every detail of hull and underwater appendages must be included, especially the grousers.

To repeat the description given in Paragraph 4.2.1, the model used in these tests was complete in every detail. Attention is again called to the photographs beginning with Figure 4-1. Chrysler has been well aware during this test program of the dangers enumerated above, and has devoted considerable effort to a testing program of thorough reliability. Reference is also made to Paragraph 4.2.18 where the reliability of predictions based on tests of a fully simulated model is explored.

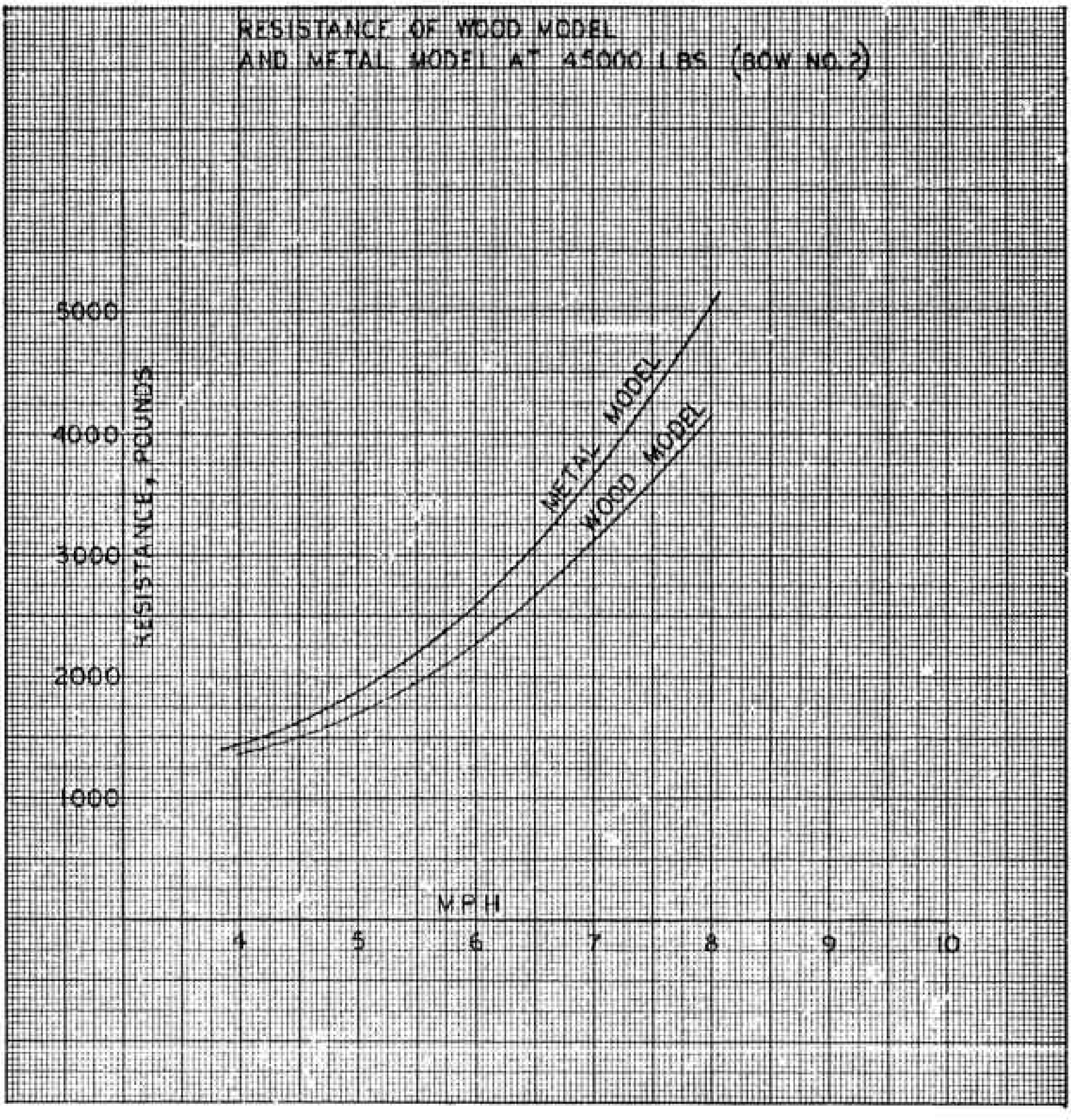

Figure 4-16 Comparison of Wood and Metal Model

4.2.4 Experiments Concerning Tracks. As shown in Section 6.2, Appendix A, there would be good reason to expect that increasing the transverse length of the grousers would result in greater area of the slip stream and therefore greater thrust. Accordingly, grousers #3, #4, and #5 were built, each with the same vertical height as grouser #1 but twenty percent longer transversely. Particulars of the five patterns are given for the model. For prototype height and length, multiply by 4.5.

The grousers are illustrated in Figures 4-17 and 4-18.

GROUSER NUMBER	ENTRANCE ANGLE	EXIT ANGLE	HEIGHT INCHES	WIDTH INCHES
1	42.5 Degree	90 Degree	.94	1.12
2	42.5 Degree	90 Degree	2.34	1.12
3	42.5 Degree	90 Degree	.94	1.25
4	35 Degree	90 Degree	.94	1.25
5	42.5 Degree	97.5 Degree	.94	1.25

Grouser #2 was the same as #1 except for height. Its shape alongside #1 is shown in Figure 4-17 and 4-18. Increasing the height, without increasing the width, does not increase the efficiency of the grouser in imparting momentum to the water because the frictional loss is high.

Note that grousers #1, #4, and #5 were all the same height. The comparison of grousers #1, #4, and #5 are shown in Figure 4-19.

The wider grousers did not do better. There must be some good reason, but it is not readily supplied. One conjecture is that the obstructions in the suspension system can only admit a fixed rate of flow to the tracks at the

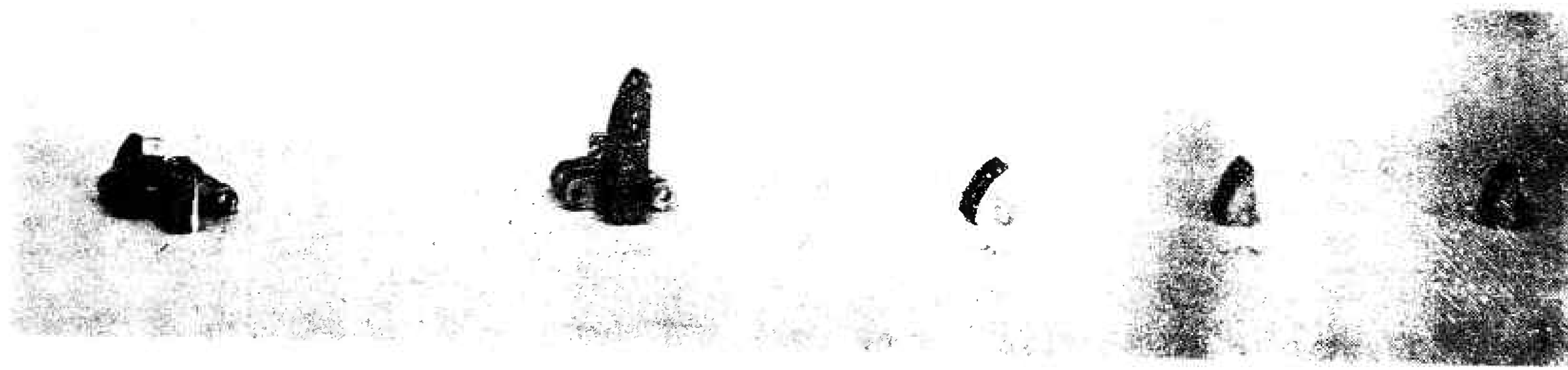

Figure 4-17 Five Grousers Tested - Side View

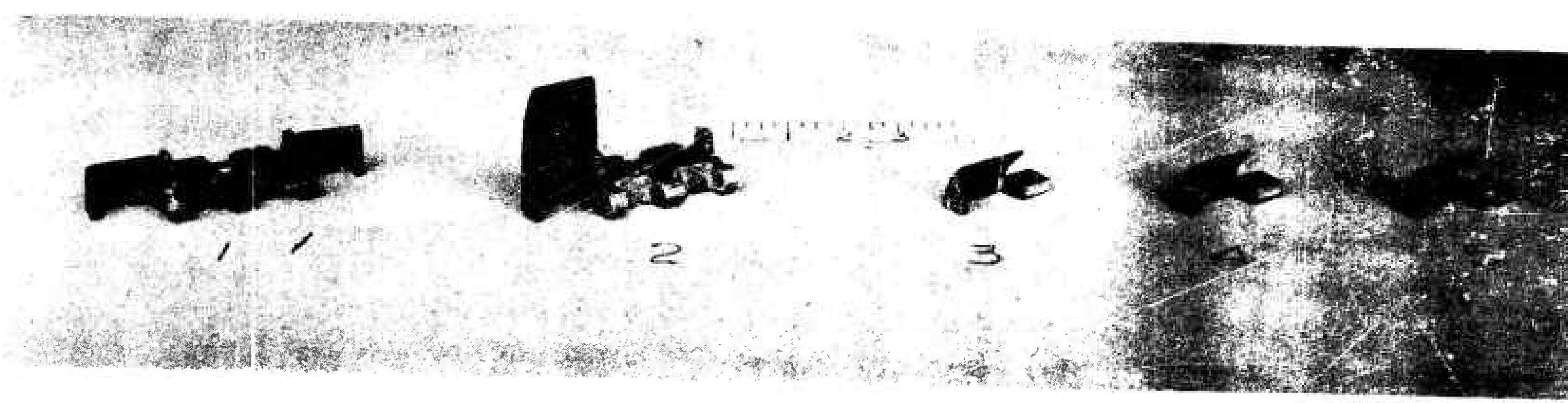

Figure 4-18 Five Grousers Tested - Angle View

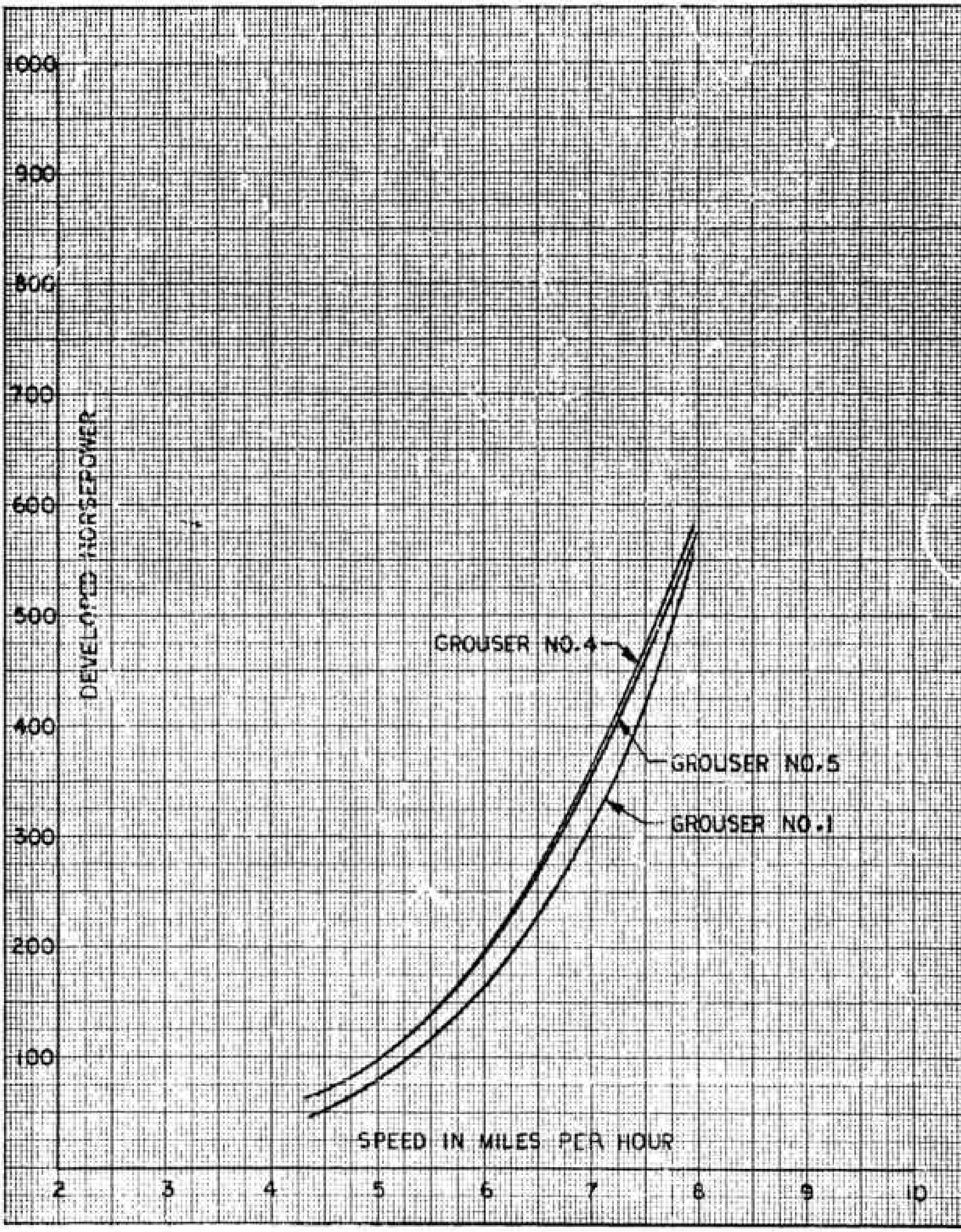

Figure 4-19 Effect Angles of Entrance and Exit of Wide Grousers

available pressure, and that a larger pump cannot increase the flow. Further investigation of this question is suggested in Paragraph 4.7.

All these grousers had fences, or flanges, at the outside edges to inhibit tip losses. (See Figures 4-17 and 4-18). (See Appendix A, Section 6.2). While grouser #1 performed better than any of the others, there is no evidence here that grouser #1 is the best of all grousers. The comparison of #1 and #2 (See Figure 4-20) does not prove that some intermediate height, nor some height even lower than #1, might not have done slightly better. It is likely that a small improvement can yet be made in propulsive efficiency by changes in grouser geometry.

4.2.5 The Return Tracks. As is evident in Appendix A, Section 6.2, the return tracks eject a stream of water in the wrong direction and also rob the lower tracks of pressure. Anything that could be done to stop this would help the efficiency. In experiments on the LVTP5 and other amphibians, restriction of the space above the return track and blockage of flow by bow fenders were both found to help. In the tests on the LVTPX12 stern baffles, to cut off the entry of water at the rear, definitely helped, (See Figure 4-21) windows cut into the side skirts did no good,(See Figure 4-22) plates below the return track hurt rather than helped.

Everybody who watches the model in the tank has suggestions. One recurring idea is to install stator vanes above the return track. These supposedly would catch the forward stream and deflect it rearward -- a change of momentum through 180 degree. The stator idea did not have much appeal because:

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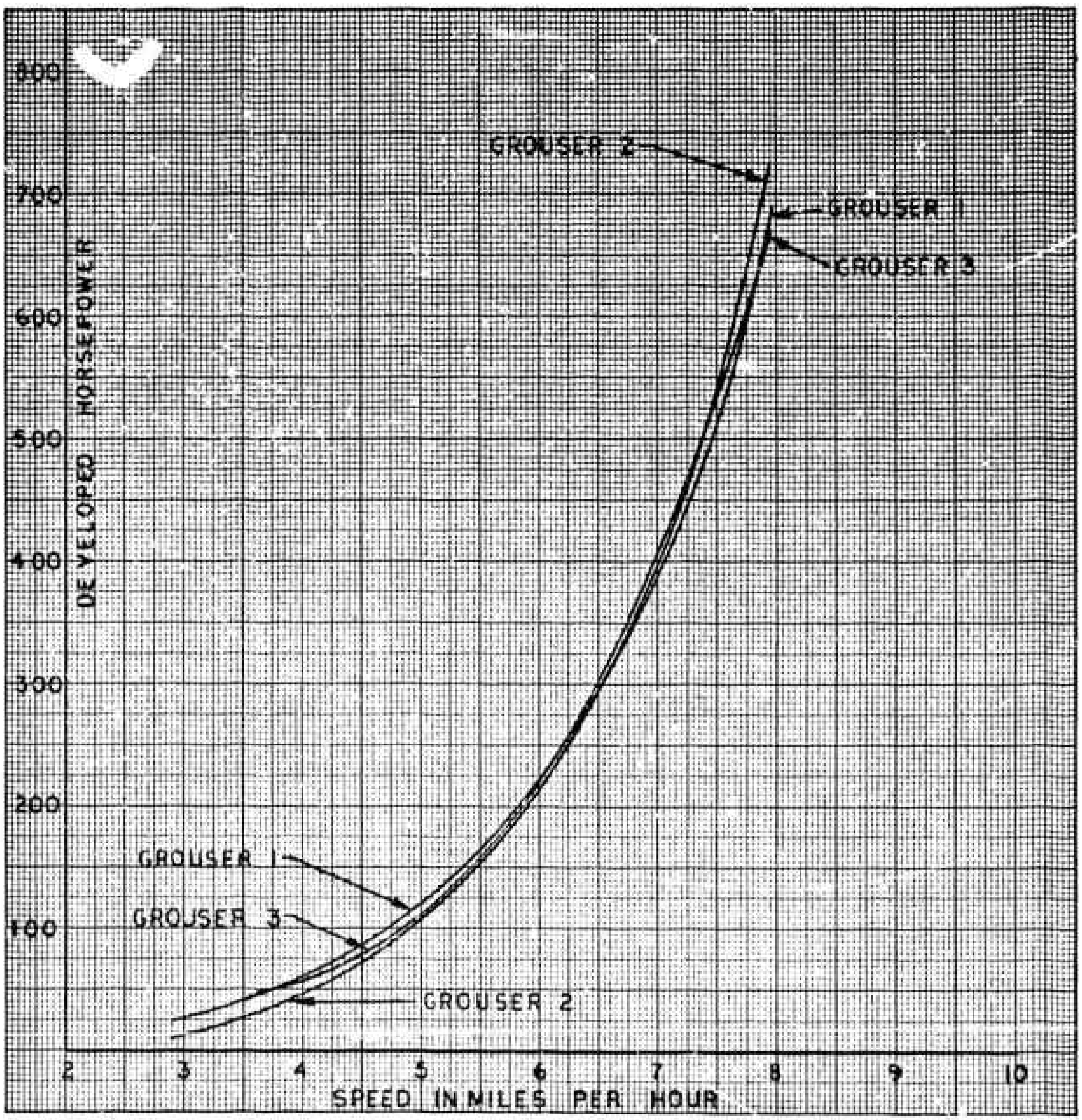


Figure 4-20 Comparison of Grousers #1, #2, and #3

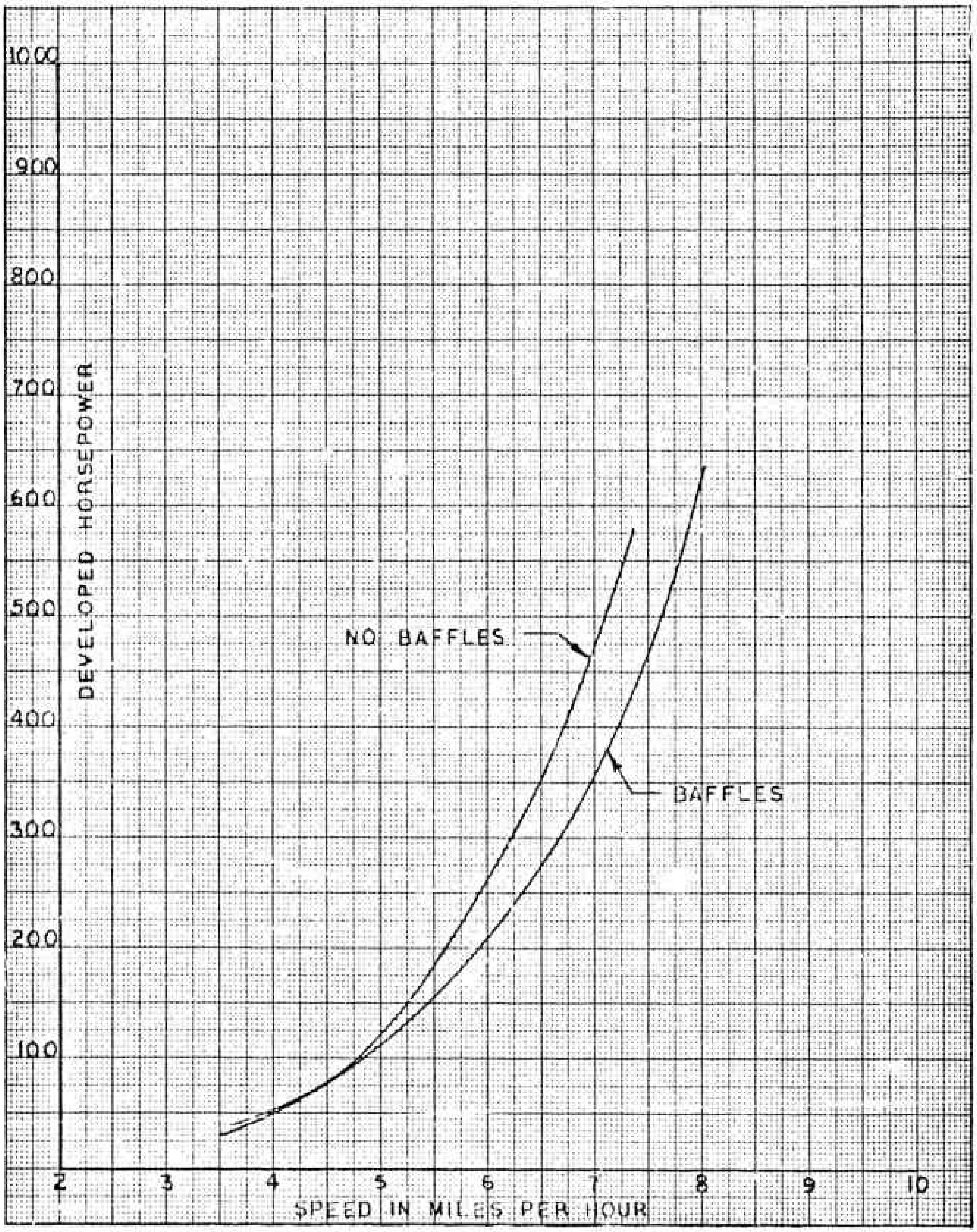


Figure 4-[illegible] Effect of Stern Baffles

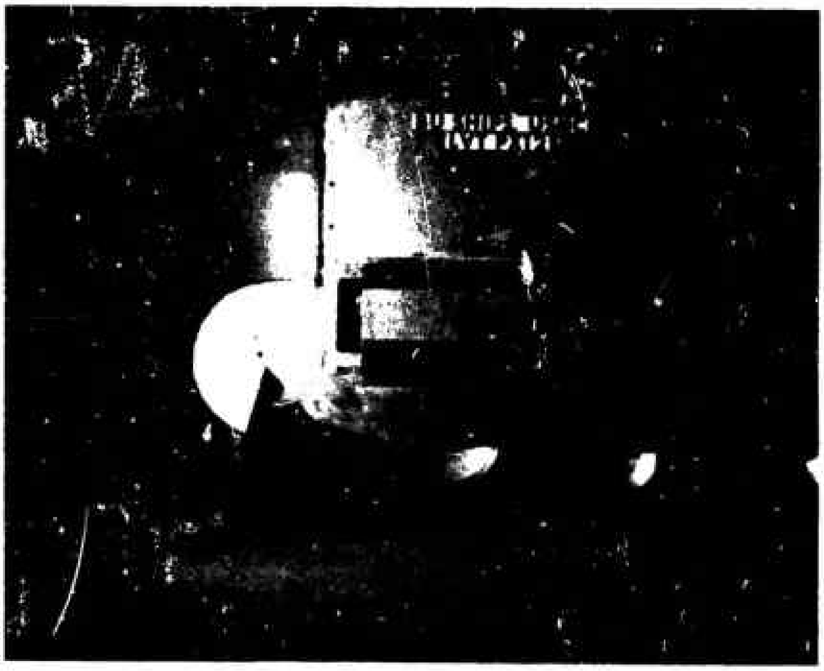


Figure 4-22 Method of Opening Windows in Side Skirts

- Even with perfect angle of attack and perfect geometry of a stator, hydraulic losses would prevent the recovery of any significant velocity.
- The stators would cake up with mud or ice.

Figure 4-23 shows the difference between generous bow fenders and no bow fenders. (Illustrated in Figure 4-3). On this same figure is shown the difference between fenders with 150 degree return and only 90 degrees. This is not the first time that fenders with 150 degrees return have been found necessary for maximum efficiency. Reports on the LVTP6 and the LVTPXII also show the same results.

Stern baffles, merely plates between body and side skirts, closing the entry of water at the rear sprocket, have a decided effect, as shown in Figure 4-24.

Does the return track build up pressure in the neighborhood of the front idler? If so, openings in the side skirts might allow the water to escape out the side instead of ejecting out the front. Adjustable windows, illustrated in the photographs (Figure 4-22), rather decreased efficiency than increased it, as shown in the graphs, Figure 4-25. Several openings were tried.

If the return track is picking up water and carrying it forward, then cutting off the water supply entirely ought to be a good idea. Choker plates lying just under the return track through the full length, fitted between the body and the side skirts, not only resulted in no benefit, but caused the craft to suck down very rapidly at high speeds. The sinkage was so bad that the

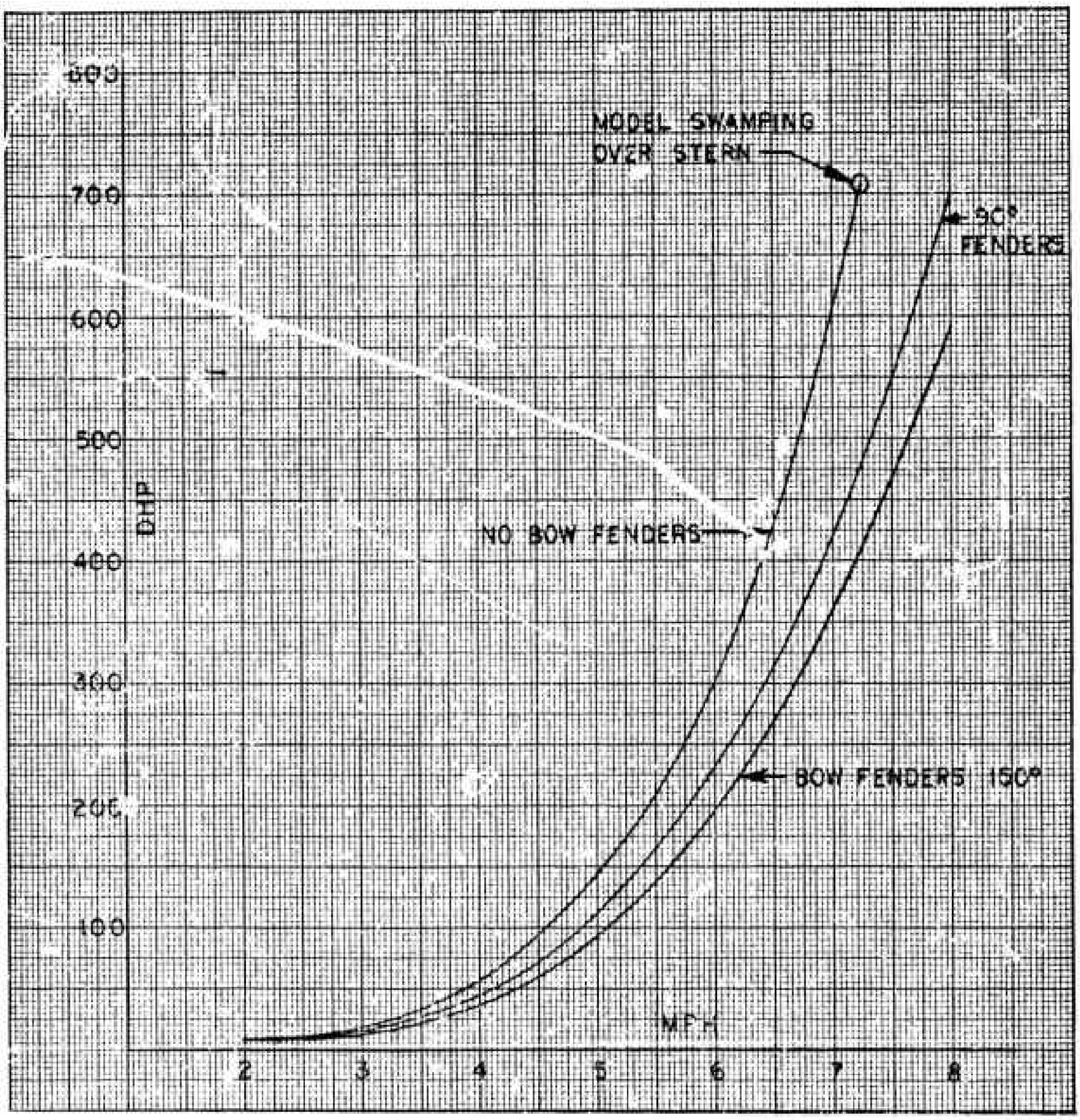


Figure 4-23 Effect of Bow Fenders

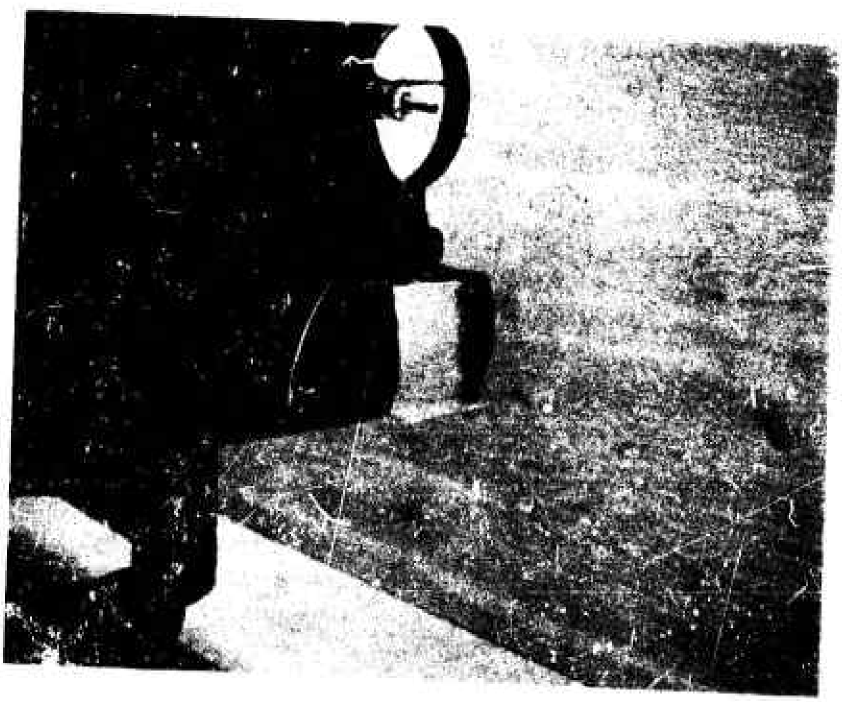


Figure 4-24 Stern Baffle, Slightly Open on Inside

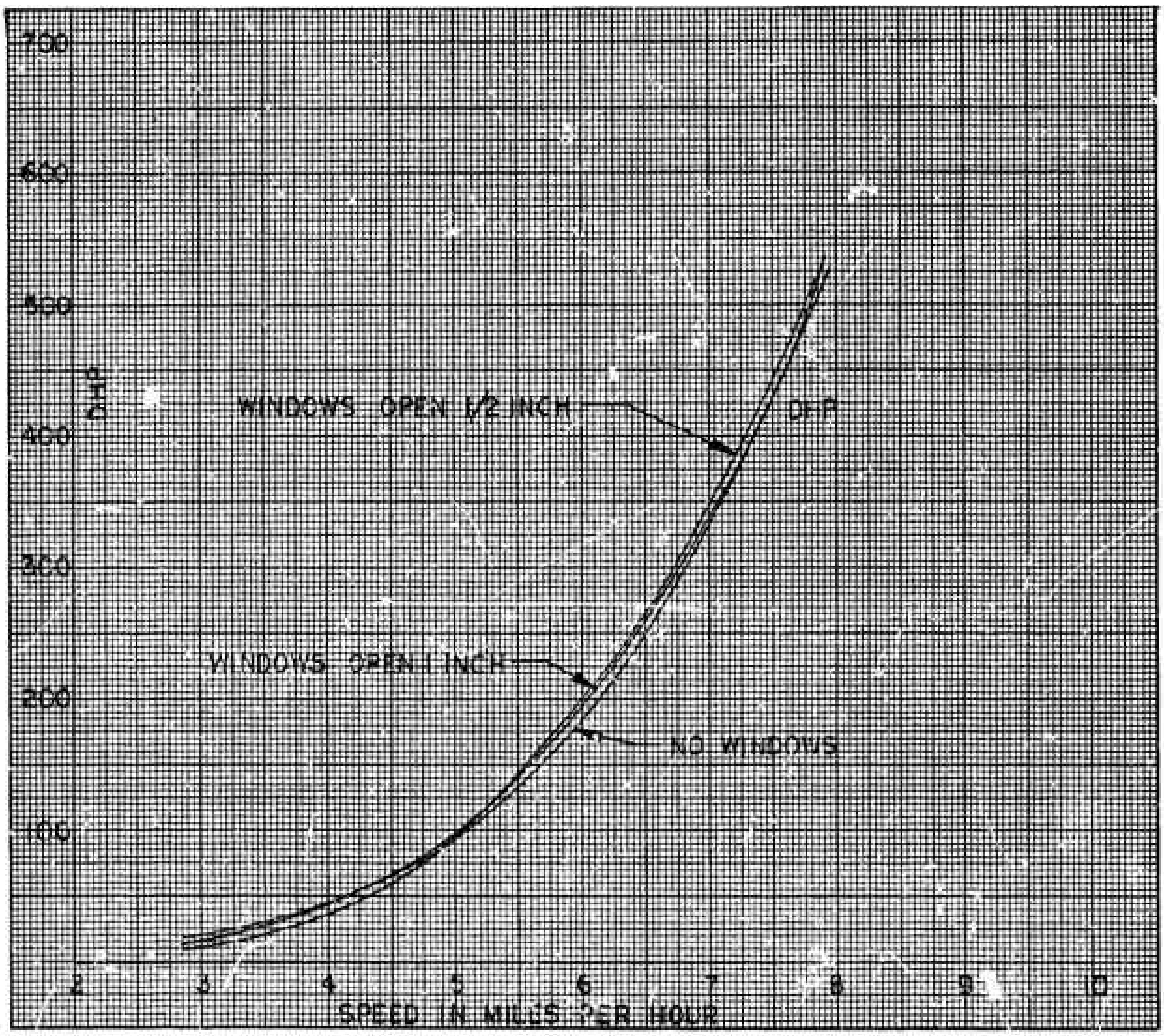


Figure 4-25 Effect of Windows in Side Skirts

test had to be stopped to prevent swamping the model. Comparative curves are shown in Figure 4-26.

Generous bow fenders, 150 degrees or more, and stern baffles do produce increased efficiency. As reported in many previous Investigations, effective bow fenders must wrap around the front sprockets somewhere close to 150 degrees. Illustrations are in Figure 4-3. The graphs in Figure 4-23 show definitely that these bow fenders are indispensible if maximum speed with track propulsion is to be attained. Figure 4-23 also shows that partial fenders are not good enough.

At the same time, the bow fenders do offer increased resistance. Figure 4-27 is from propeller tests, made to determine whether or not it would be better to place bow fenders on a propeller-driven vehicle. The results show that when propellers are used, the LVTPX12 will do better without the bow fenders. (These curves are from tests with model propellers and are <u>not</u> curves of propeller horsepower for the prototype.)

4.2.6 <u>Effect of Emergence of Return Tracks</u>. On the earliest tracked amphibians, the tracks returned over the hull, entirely out of the water. These early vehicles achieved good speed. Was this success due to the absence of resistance on the return tracks, or was it chiefly due to the light weight of these vehicles? An approximation of eliminating the water resistance on the return tracks can be obtained from the experiments in which the water was forced out of the space around the return track between the side skirts and the hull. For this experiment the side skirts, the bow fenders, and the stern baffles were sealed off tightly, while air from a vacuum cleaner pump was forced into the space thus sealed. The return tracks were thus en-

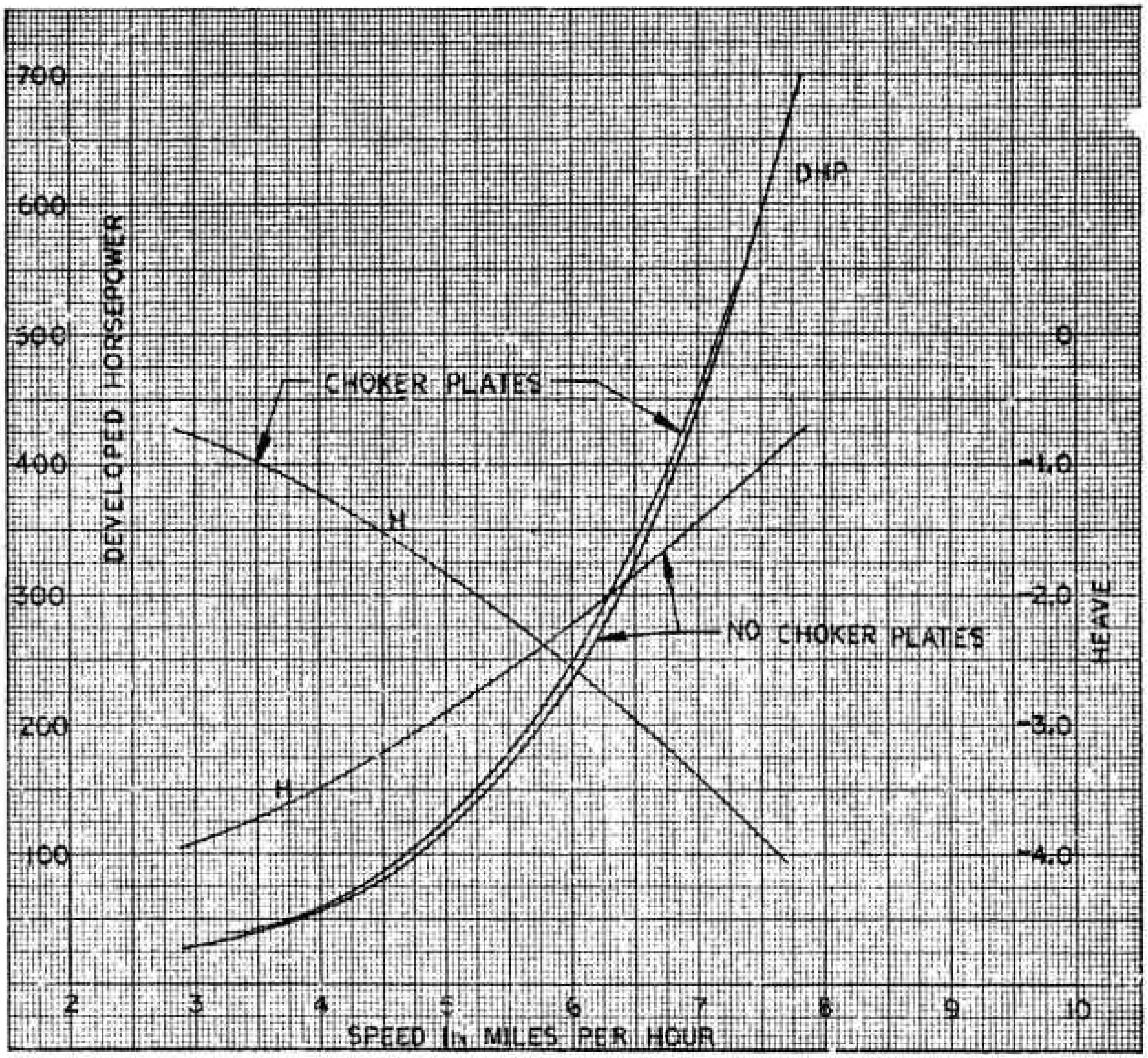


Figure 4-26 Choker Plates (Note Effect on Heave)

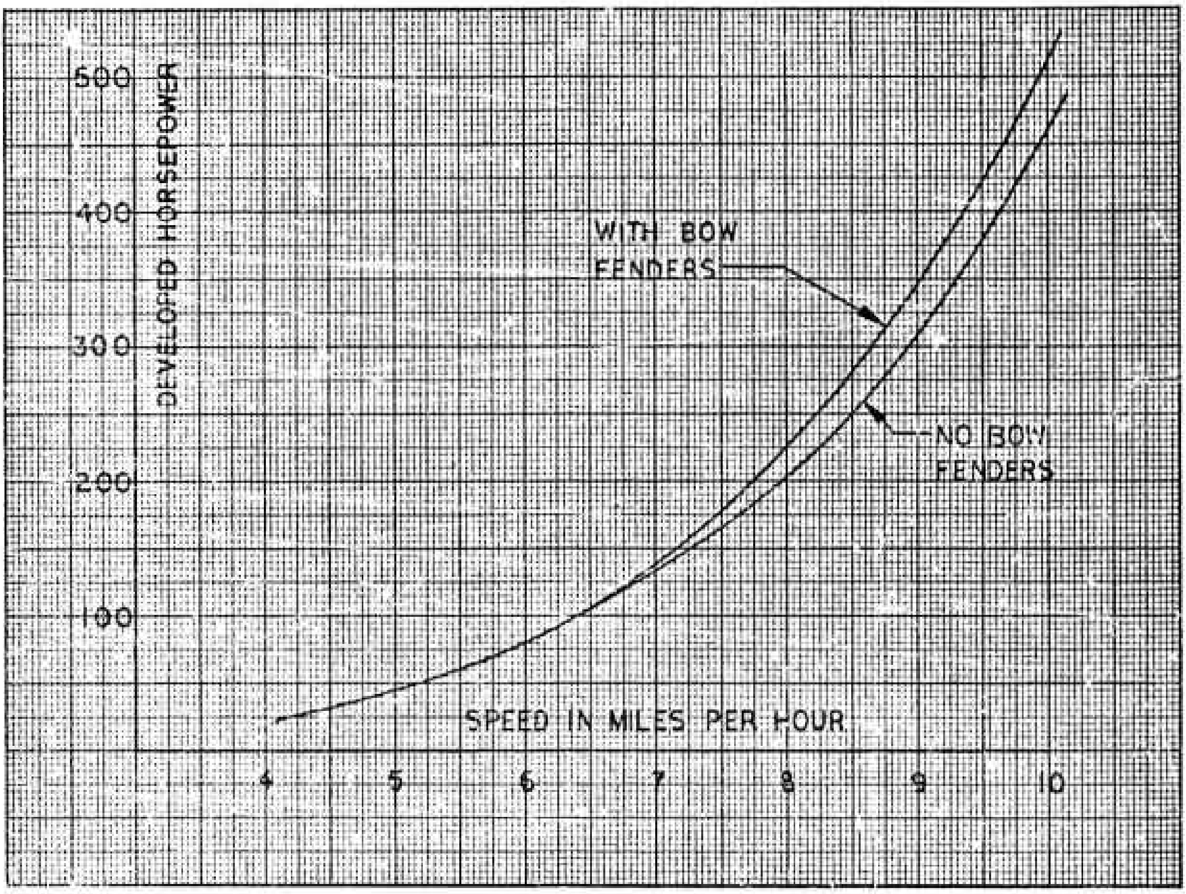


Figure 4-27 Increase of Horsepower Due to Bow Fenders - Screw Propulsion

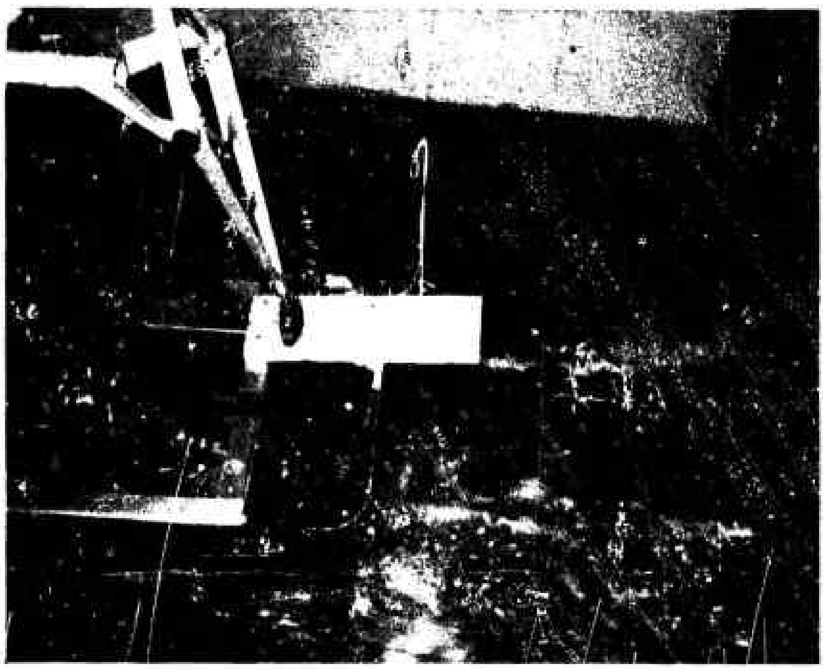


Figure 4-28 Rear View of Model with Air Hose

tirely in air. The air hose leading to the sponson is illustrated in Figure 4-28. When the model was under way, a large amount of air was carried by the grousers, and the water around the model became quite milky. The results of this experiment, compared with the identical model when no air was introduced, are shown in Figure 4-29. Although at low speeds some improvement results, at high speeds the difference should be disregarded. Although the weight of the model is not decreased by this air, the draft is decreased. The virtual weight also is decreased, because not as much water is being carried along under the sponson. At high speeds, the tracks carry the air out as fast as it can be pumped in, so that the curves come together. At the same time, the entrained air around the tracks at high speed lowers their efficiency. From these tests, the evidence is that at some speeds the LVT would do better with the tracks returning in air, but not necessarily at all speeds. It cannot be concluded from this experiment whether the better performance at lower speeds was due to the air around the return tracks or due to the decreased virtual weight.

4.2.7 Stern Planes, Contravanes, and Stern Baffles. Early in the test program, a broad plate flush with the bottom, extending horizontally rearward from the transom and outward to the outside of the tracks, produced a jump in propulsive efficiency. (Figure 4-30 illustrates the stern plate). Behavior with and without the stern plate is shown in Figure 4-31. The improvement was in spite of the obvious increase in the amount of water dragged along in the wake. The phenomenon did not agree with reasoning, except for the possibility that the plate was acting as a stripper, deflecting the stream from the tracks horizonally.

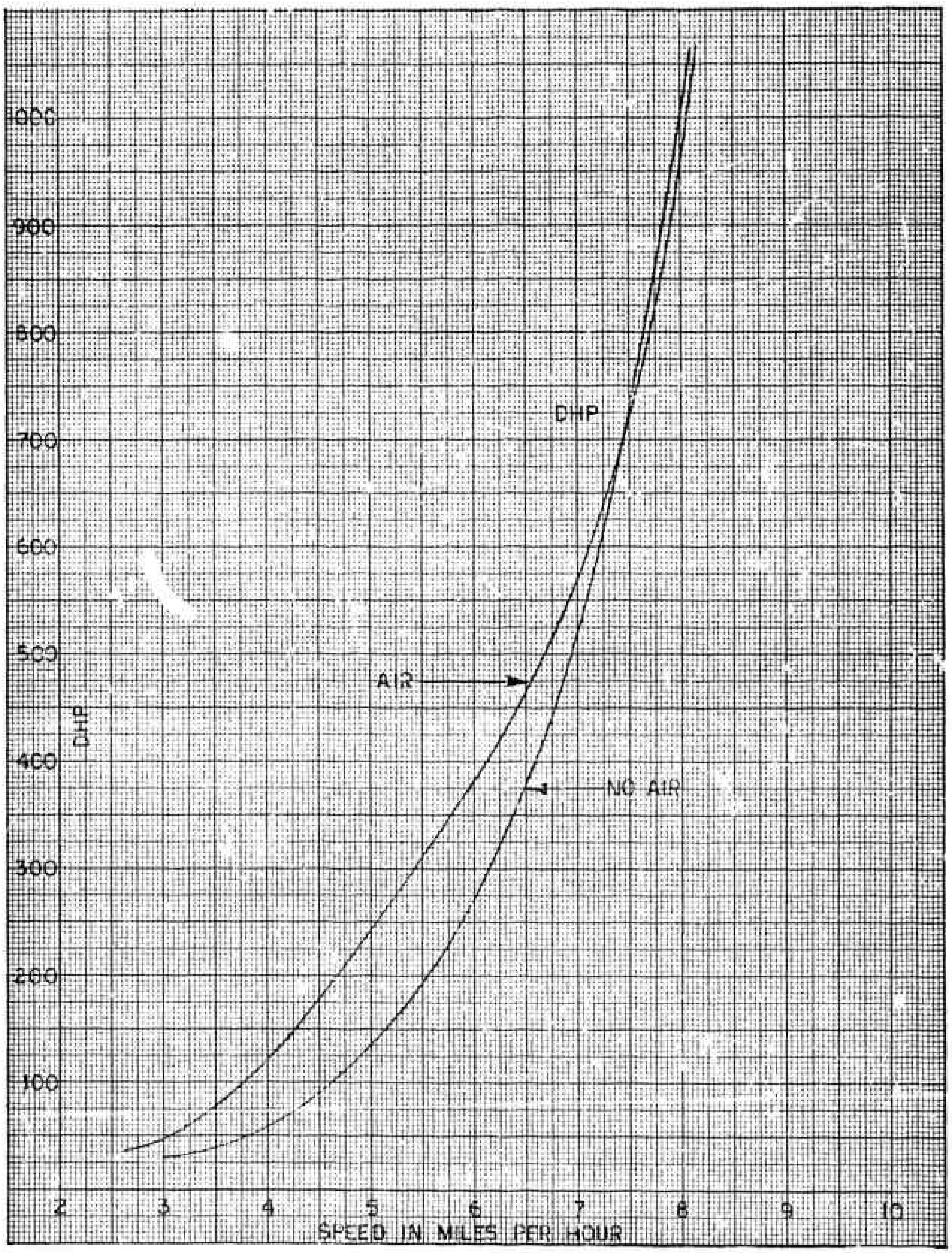


Figure 4-29 [illegible] with Air

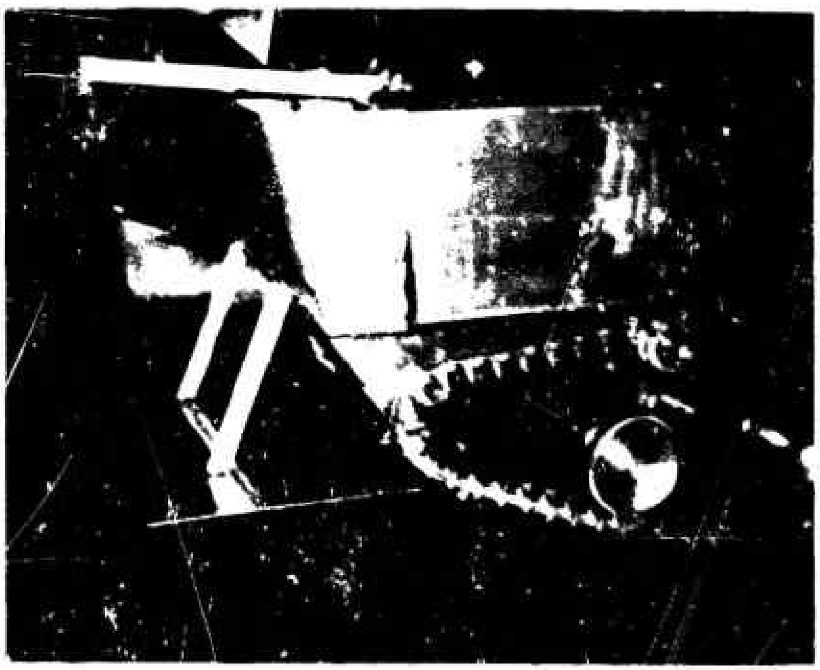


Figure 4-30 Stern Plate

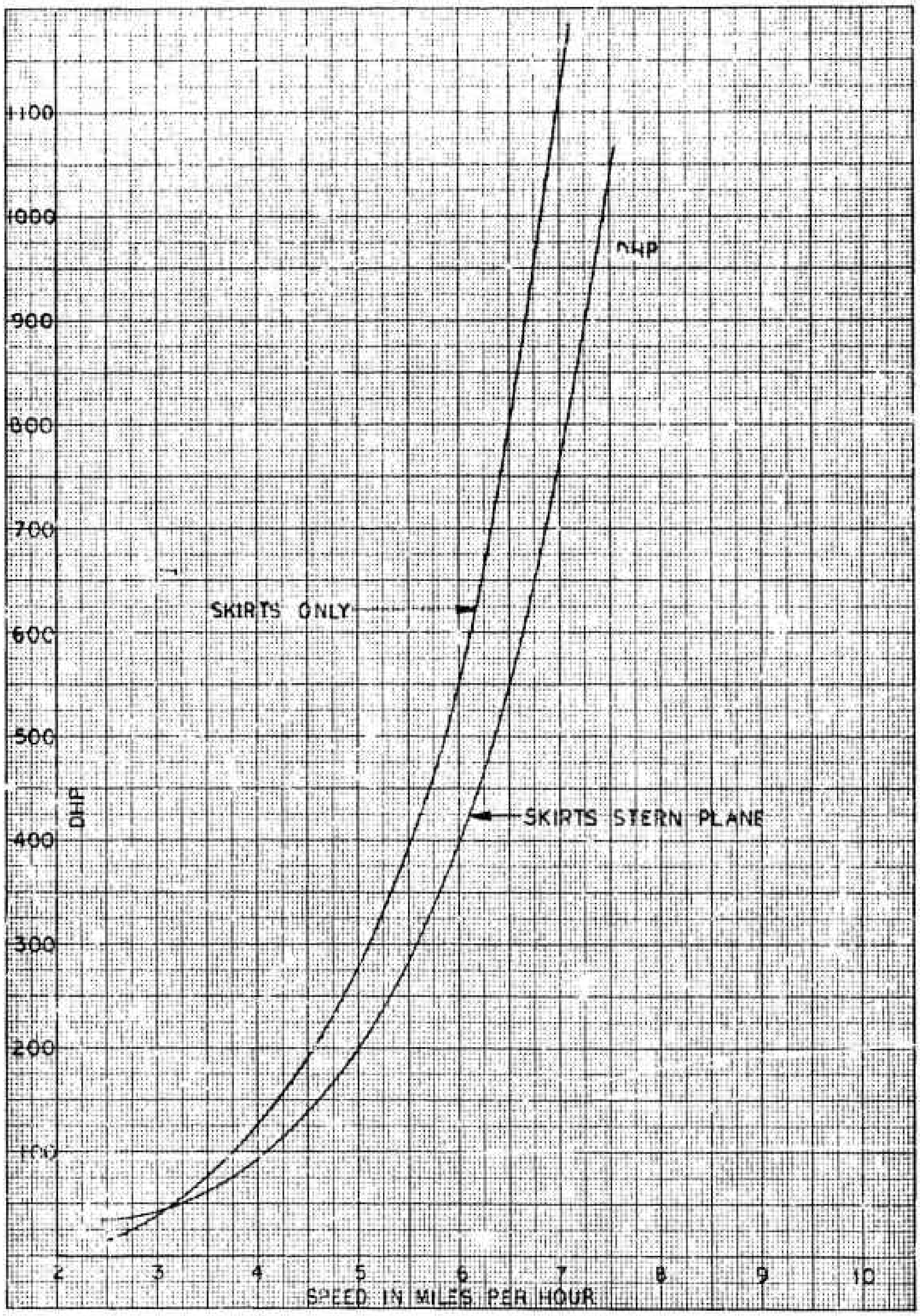


Figure 4-31 Effect of Stern Plate

This effect would be similar to the principle employed by Arthur Rigg, British inventor, in his application of contravanes to paddle wheels in 1860. (See Reference I, Appendix A, Section I.0). Rigg's contravanes improved the performance of paddle wheels, and they have also markedly improved the performance of tracks. The contravanes are illustrated in Figure 4-32. Figure 4-33 shows the difference between contravanes and the stern plate. The elevation, angle, and size, of the contravanes, however, is critical. Figure 4-34 shows several curves for successive positions of the contravanes. Moreover, the optimum position and angle of the contravanes is dependent on the initial trim of the craft. The force of water against the vanes can raise the stern enough at high speeds to cause the vehicle to trim down by the bow. Since no single position or angle is optimum for all initial trim conditions, attainment of maximum benefit would demand that the contravanes be designed with some adjustability.

In Figure 4-32 the contravanes are shown with lips on the sides, intended to prevent the water spilling over from the high pressure to the low pressure side of the plate. Experiments with contravanes having no lips show no practical difference, according to the curves in Figure 4-35

Test No. 34 (See curve in Figure 4-36) showed the highest propulsive efficiency for any of the experiments on the track propelled model. But in this test the contravanes were very long, 4.5 feet on the prototype, This length is obviously objectionable. After 22 1/2 inches of the contravanes were cut off, the model performed not quite as well, but almost as well. Test No. 34 is for the model at 45,000 pounds prototype weight, with 4.5 foot contravanes, while test No. 54 is for 45,000 pounds and 2.6 foot contravanes.

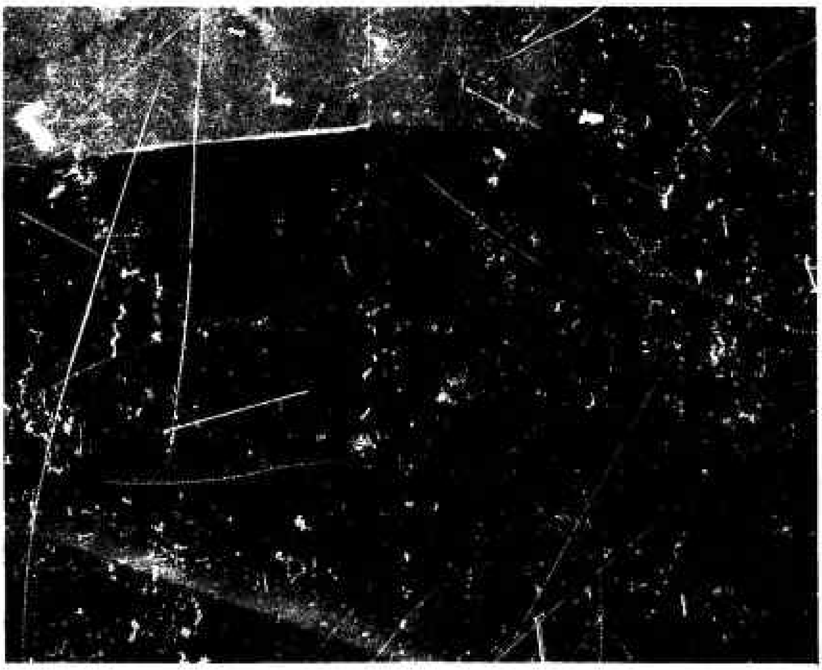


Figure 4-32 Contravane, 2.6 Feet Long (Prototype Scale)

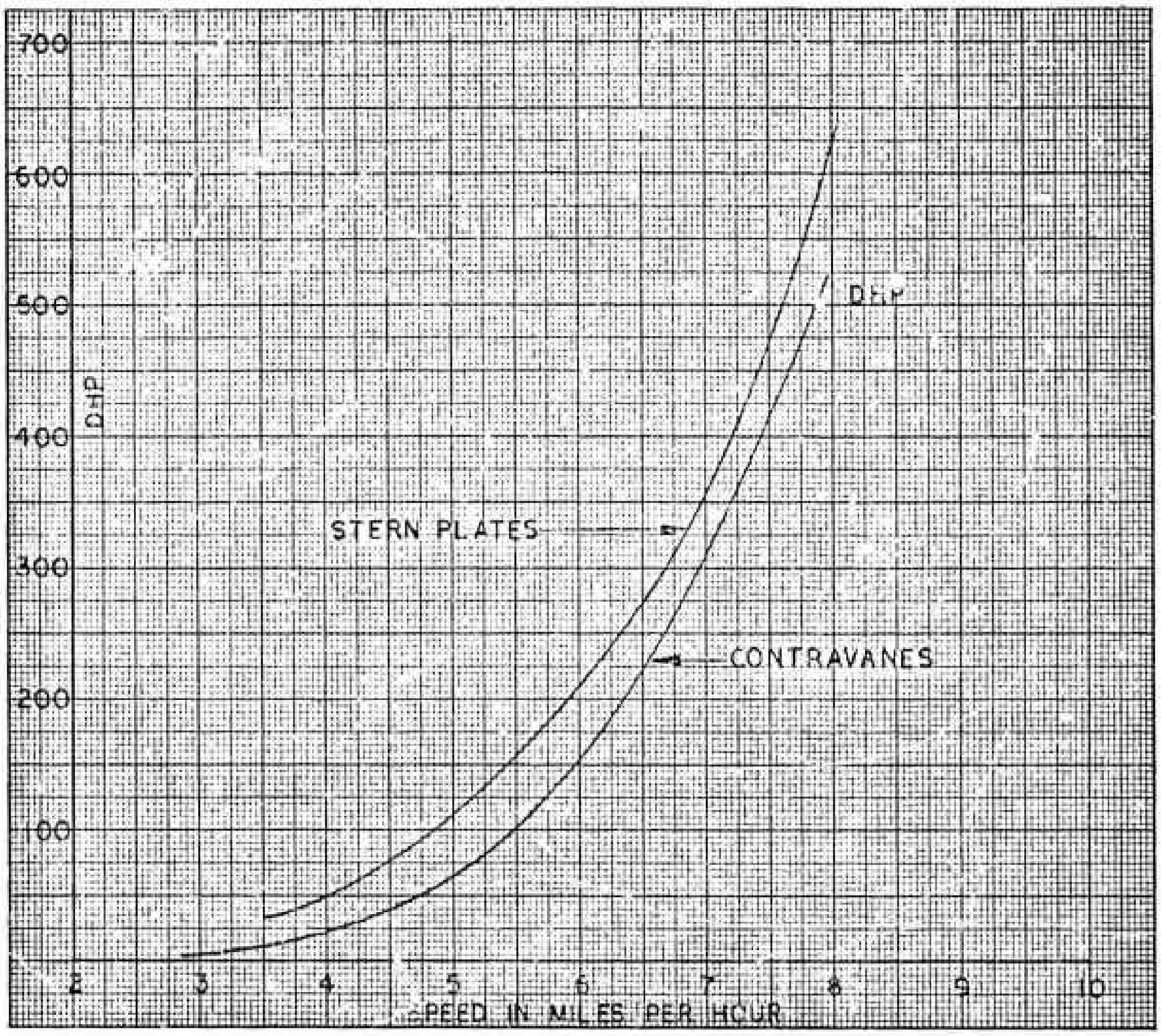


Figure 4-33 Comparison of Stern Plate and Contravanes

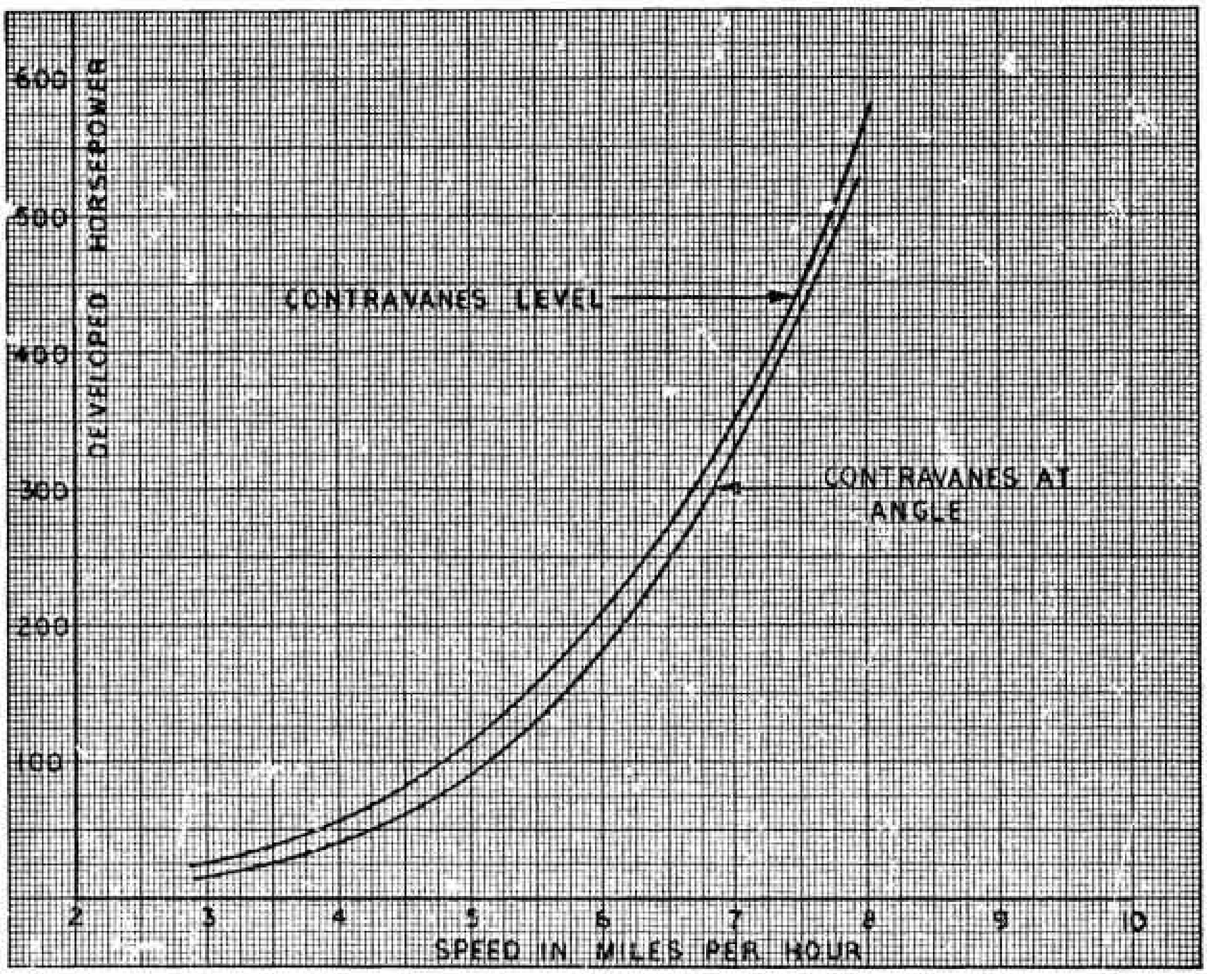


Figure 4-34 Two Tests with Contravanes

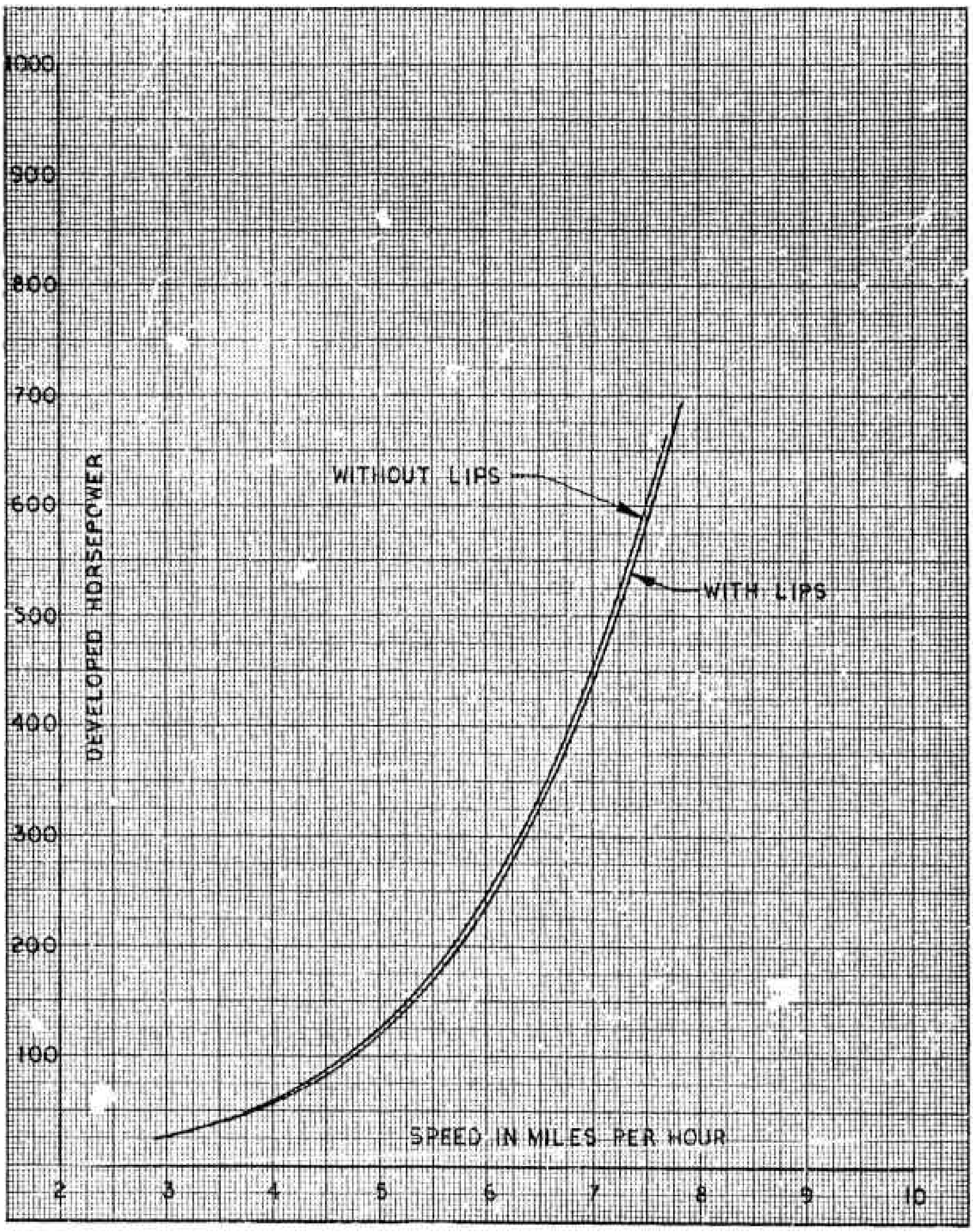


Figure 4-35 Effect of Lips on the Sides of Contravanes

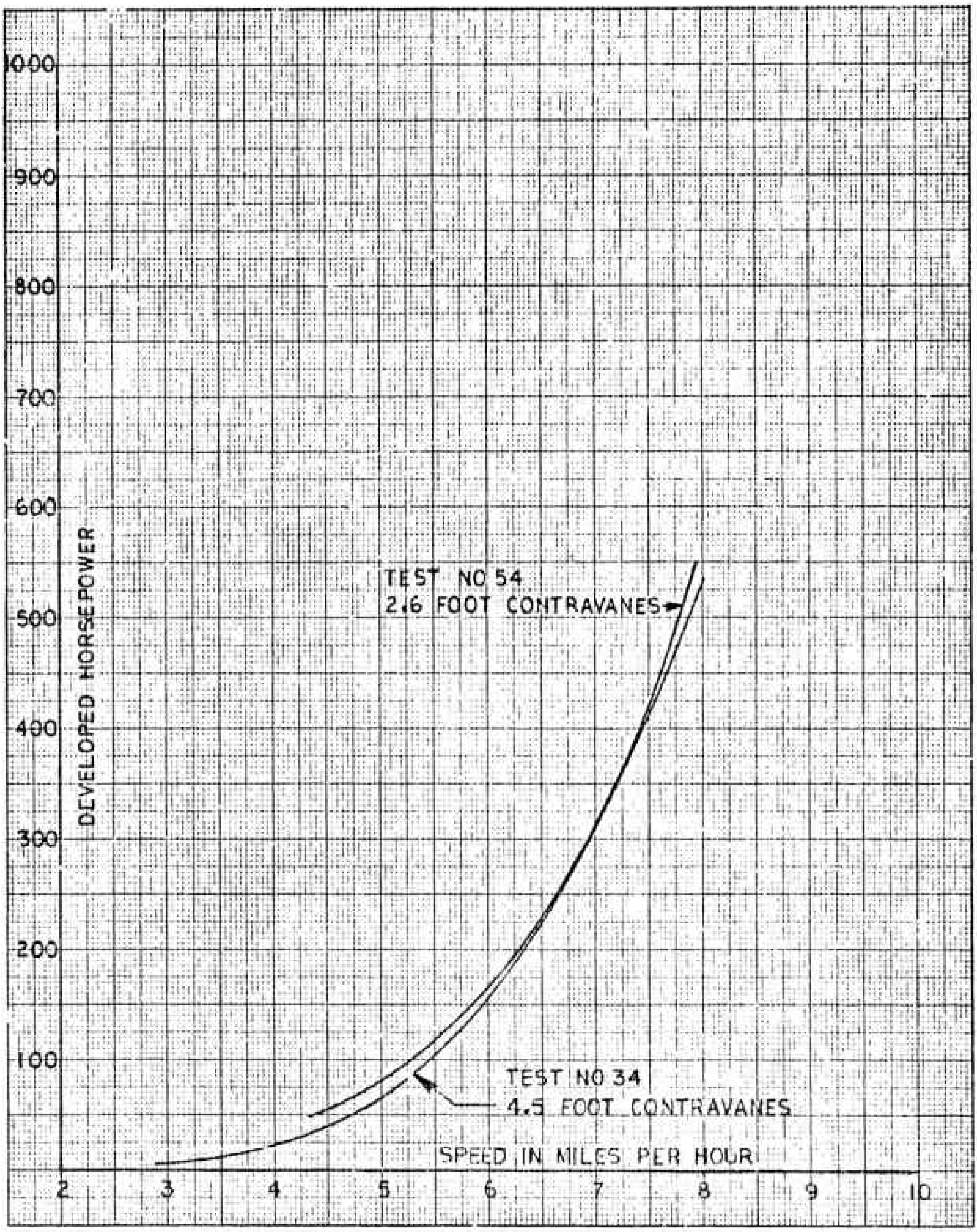


Figure 4-36 DHP at 45,000 Pounds, Test No. 34 and Test No. 54, Both with Bow #3

In all cases, for maximum efficiency the rear sprocket must be closed off by a stern baffle. A modification of the stern baffle, illustrated in Figure 4-24, produced efficiency as good as the short contravane with baffle. The baffle illustrated in Figure 4-24 should fit tightly over the rear sprocket. At the level of the bottom of the vehicle it has a horizontal contravane 4 inches long. This device will be more acceptable from an operational standpoint than either the stern plate or the contravane. Its effect is as good as that of the contravane, as shown by the curves in Figure 4-37. (In interpreting the results of model tests, extremely slight differences should not be taken as definitely ranking one configuration above another. In model testing, even with the finest apparatus, two to three percent deviation in DHP may be expected from one experiment to another.)

At 8 MPH, the model trimmed down by the stern 2 degrees with the stern baffles alone, while with the contravanes the change of trim can be controlled. This means that if the vehicle is trimmed down by the bow in the static condition, it will tend to level out when under way with the stern baffles. If the vehicle is statically trimmed by the bow in most conditions, it will be better to adopt the stern baffles with the short contravanes (4 inches) and thereby eliminate the operational inconvenience of the longer contravane.

4.2.8 Bow Planes. A large plate entirely beneath the surface, angling upward from the bottom of the model and extending forward 12 inches (4.5 feet on the prototype) had a bad effect. The static pressure of the elevated bow wave, standing directly above the plate, built up much faster than the dynamic pressure on the bottom of the plate, and quickly submerged the bow. The result is hardly worth reporting, but is illustrated in Figure 4-38.

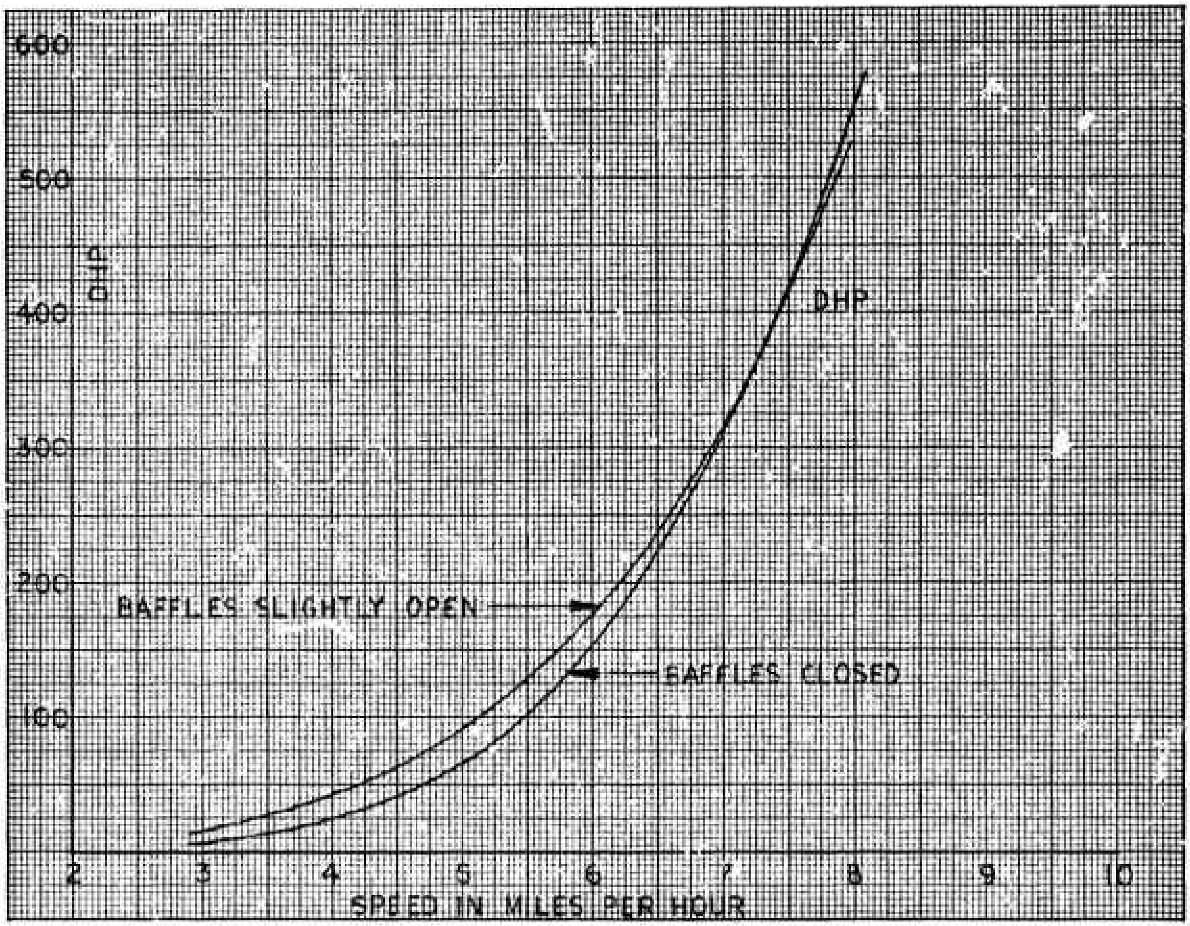


Figure 4-37 Effect of Open Stern Baffles

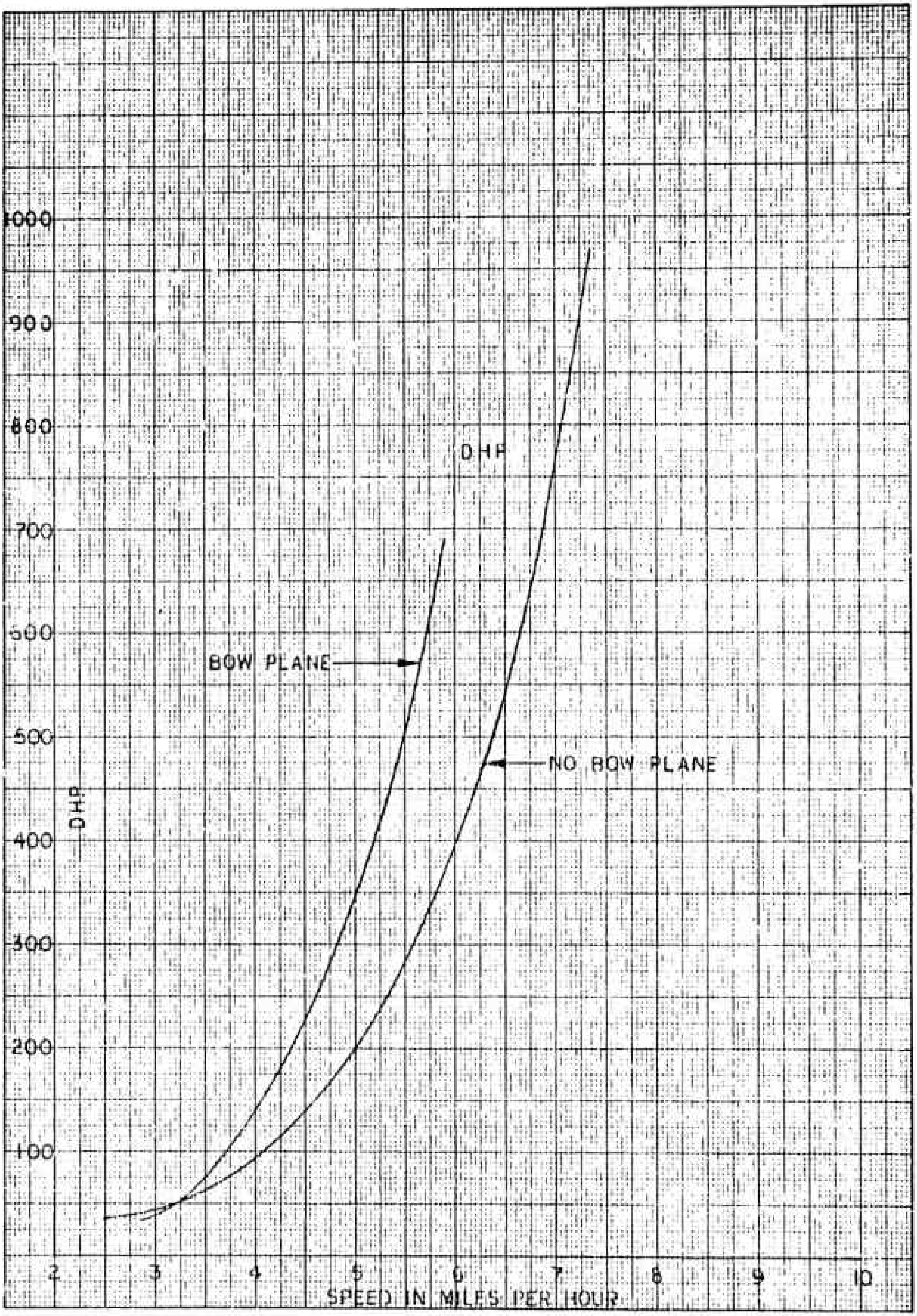


Figure 4-38 Submerged Bow Plate

A bow vane protruding above the surface was installed on the model of the LVTPX7 at David Taylor Model Basin (Reference 20, Section 1.0, Appendix A) chiefly to allow the craft to go without submergence at 10 miles per hour. On the LVTPX7 the vane knocked down the bow wave and allowed the desired speed, although it is not certain that it produced any significant reduction in resistance. A similar vane on the LVTPX12 did not help propulsive efficiency, as shown in Figure 4-39. At very high displacements, say 59,000 pounds, the submergence of the LVTPX12 does become annoying at high speeds as shown later, and if the weight goes that high some such bow vane may become worth trying. It will be difficult to accommodate any vane on a boat-shaped bow. There is no assurance that it will produce the desired results. It will not reduce the resistance.

4.2.9 Form of Ends. The several bow sections tested during the program are illustrated in Figure 4-2, 4-3, and 4-4. The photographs and legends should be self-explanatory. Bow #3 is the finest entrance that can be designed within the allowable limits. To achieve this fineness, the bow has to be extended but the stern has to be cut off so that the total length remains 26 feet. This results in a plumb transom, but the benefit of a fine bow is obvious in all the tests. The reason for this is that the moving water presses on the bow with a force proportional to some power of velocity, while the reduced pressure on the stern is a function of separation -- the change of smooth, laminar flow to violent turbulence. The separation is no worse with a plumb transom than with a stern slightly raked. Unless the craft is extended greatly in length aft with a gentle run, very little can be done to reduce the rearward force on the stern, while on the other hand the adverse force on the bow reduces

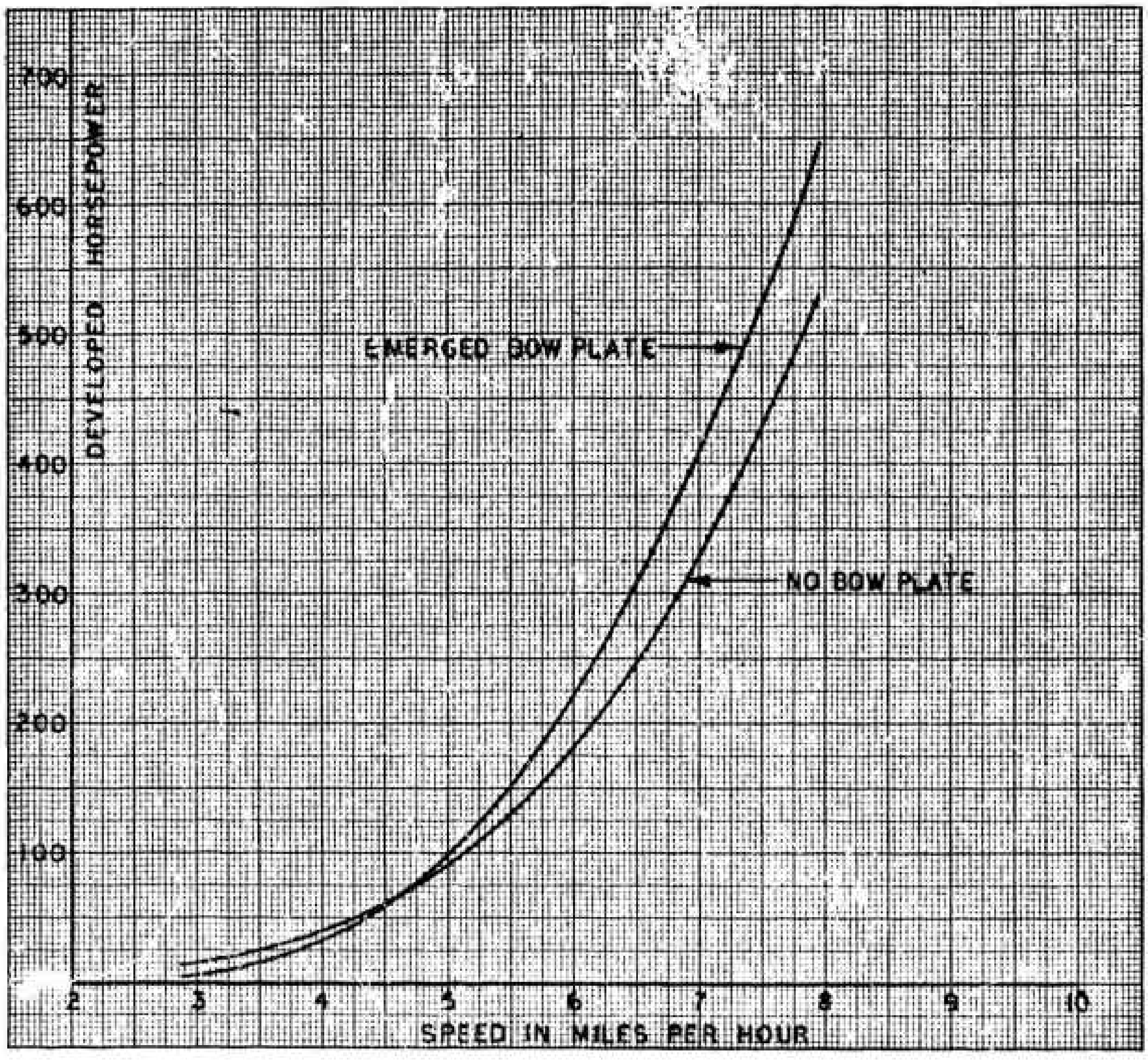


Figure 4-59 Effect of Emerged Bow Plate

rapidly with reduction in drag coefficient. Figure 4-40 illustrates the effect of fining the bow section and blunting the stern.

At the same time, the effect of stern shape is illustrated in Figure 4-41, in connection with bow section #3. The stern sections tried in the tests are shown in Figures 4-5, 4-6, 4-7, and 4-8. Tapering the stern above the sponsons produced a slight apparent improvement.

The low resistance illustrated by both curves in Figure 4-40 was obtained by covering the track grousers with tape, thus simulating a fairly smooth track which might be suitable for land propulsion. Such a track would still propel the vehicle in water at some speed, say 4 or 5 MPH, but no experiments were performed for track propulsion with smooth tracks. Although the wake fraction behind the craft is very large -- even 100 percent and more in spots -- very little can be done about it within the allowable length and the necessary displacement. Even if the length or displacement could be altered to allow fining of the stern, the finess would still be more profitably invested in the bow, up to a point much farther than was reached with bow section #3. These results illustrate a principle commonly known: if violent separation of flow is occurring at the stern of a craft, resistance will not increase if the hull is cut off at the point where the most violent separation begins. Unless the stern of the LVTPX12 can be given a very gentle run, say 15 degrees rise from the horizontal, there is no point in trying to improve it. A square stern will do as well as one with a slight rake.

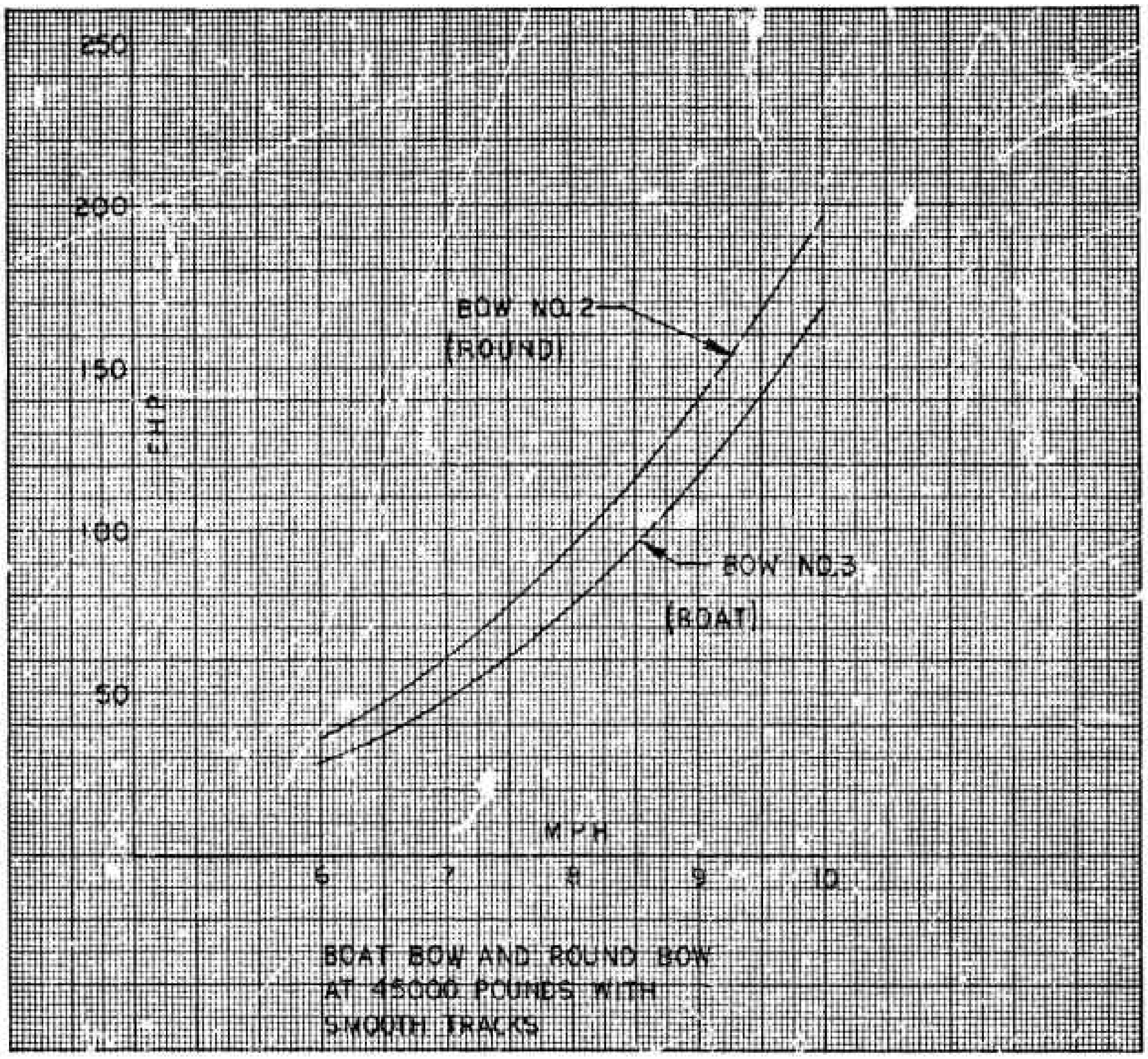


Figure 4-40 Effect of Bow Shape

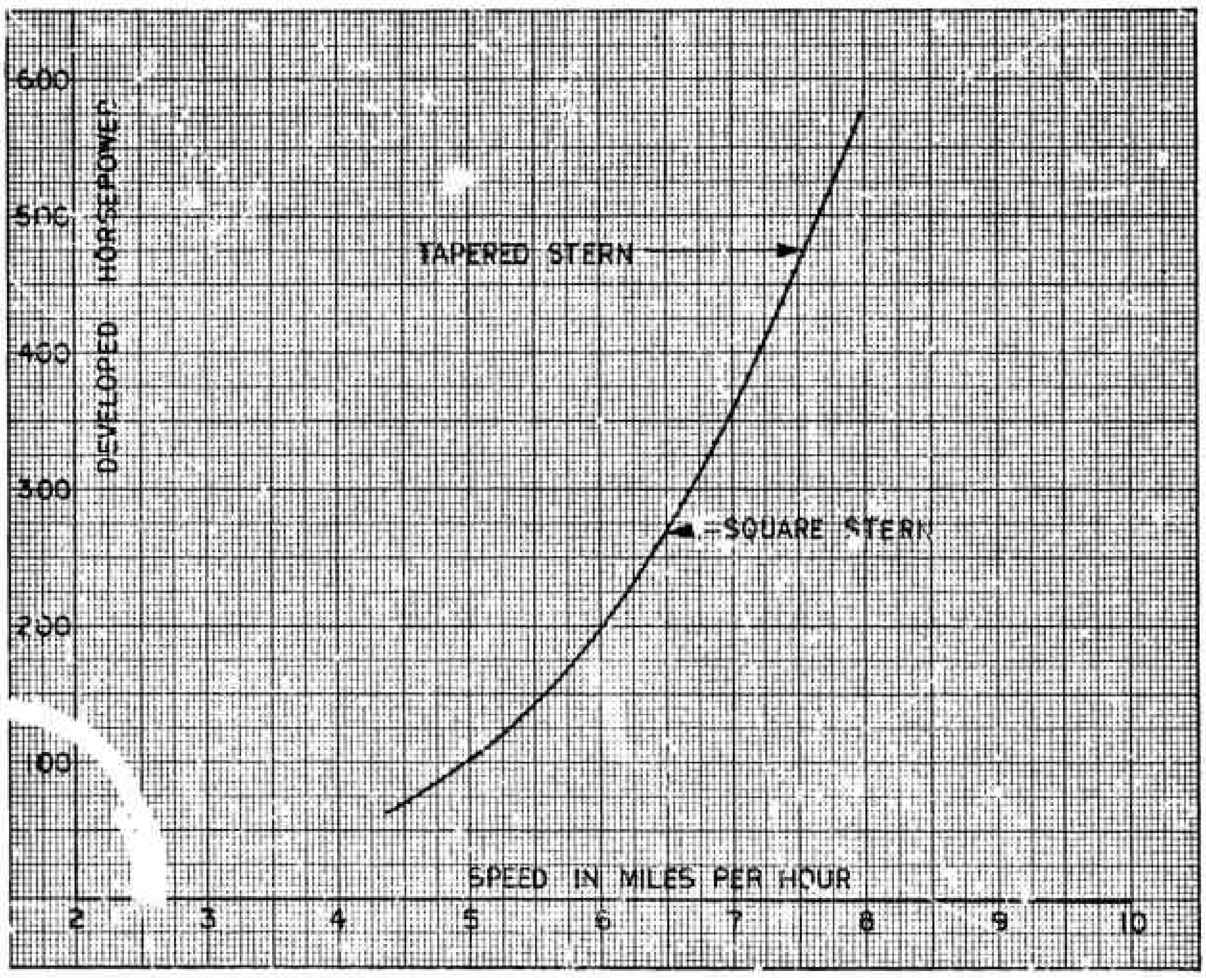


Figure 4-11 Effect of Tapering Stern (Vertical Transom)

The difference between a very flat bow and a round one is shown in Figure 4-42. These tests were both with the model in the bare condition — no side skirts nor other appendages of any kind. Hence the difference can supposedly be attributed entirely to the easy shape of bow #2, which may be compared by the photographs in Figure 4-2 and Figure 4-3.

4.2.10 <u>Effect of Beam.</u> Lest any opportunity for designing a vehicle significantly better in all respects be missed, there is found in Section 5.0 a detailed study of an LVTPX12 having 20 percent less beam then the specified 10.5 feet. The lower resistance of this narrow vehicle must be weighed against its difficulties. The first question is the extent to which resistance will be reduced. For this purpose only towing tests were performed. The wood model, after being tested for towing resistance with 10.5 foot beam, was reduced in beam by 20 percent and the tests repeated. The results are clearly shown by the curves in Figure 4-43 and Figure 4-44. In general, for this particular vehicle, reducing beam by 40 percent results in a reduction of an 8 to 9 percent in resistance. But the small reduction in resistance achieved by reducing beam does not necessarily mean that the horsepower for the track-propelled vehicle can be correspondingly reduced. Nothing in these towing tests says anything about propulsive efficiency. There is no reason to expect that the propulsive thrust of the tracks would be increased by the narrow beam, but rather that crowding the tracks closer together both vertically and horizontally would be more likely to decrease the useful thrust. To whatever extent this happened, the small decrease in resistance would be lost in reduced thrust, and the narrow vehicle would perform about the same as the wide one. An additional reservation concerning this experiment is that the model was

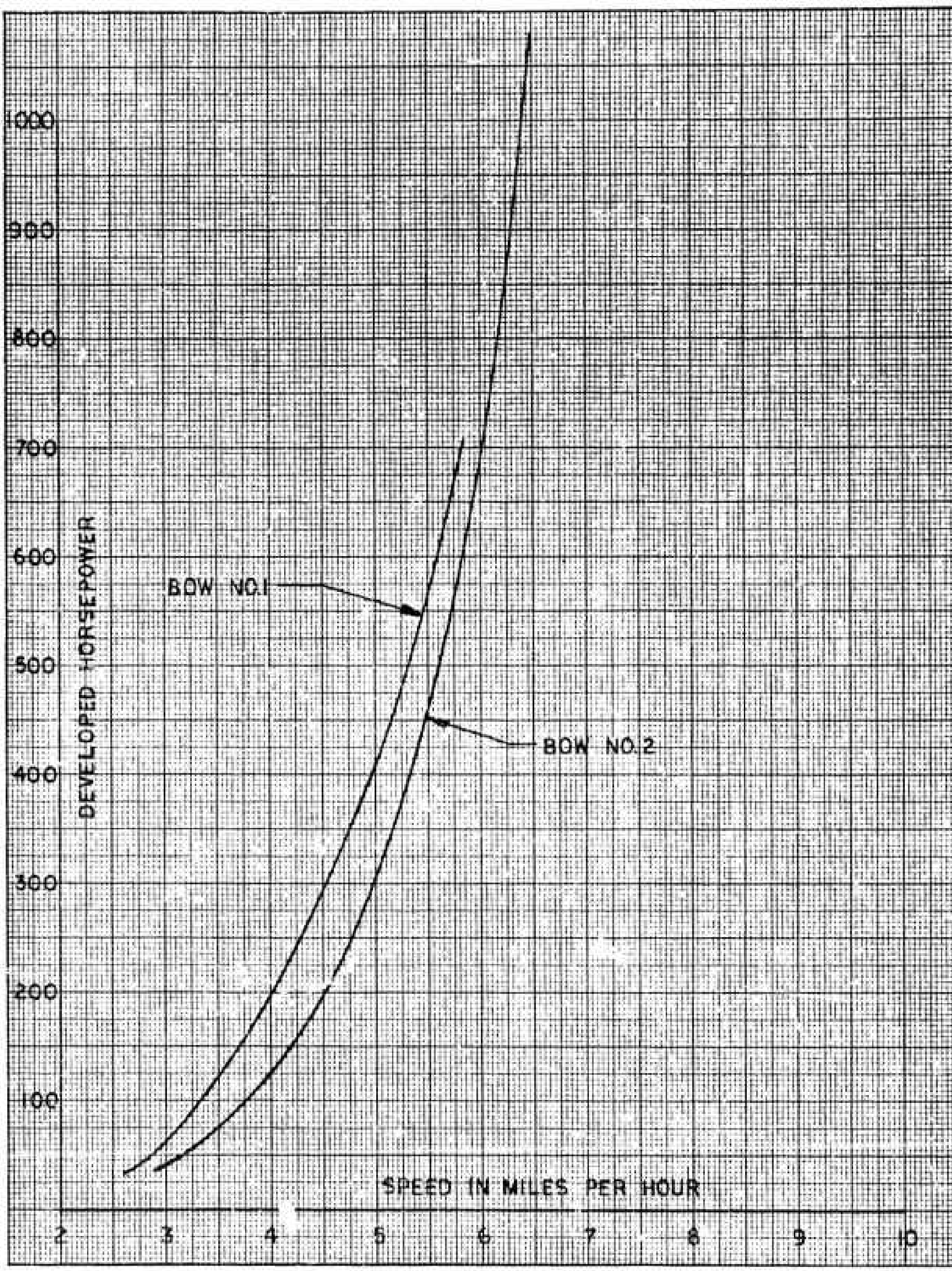
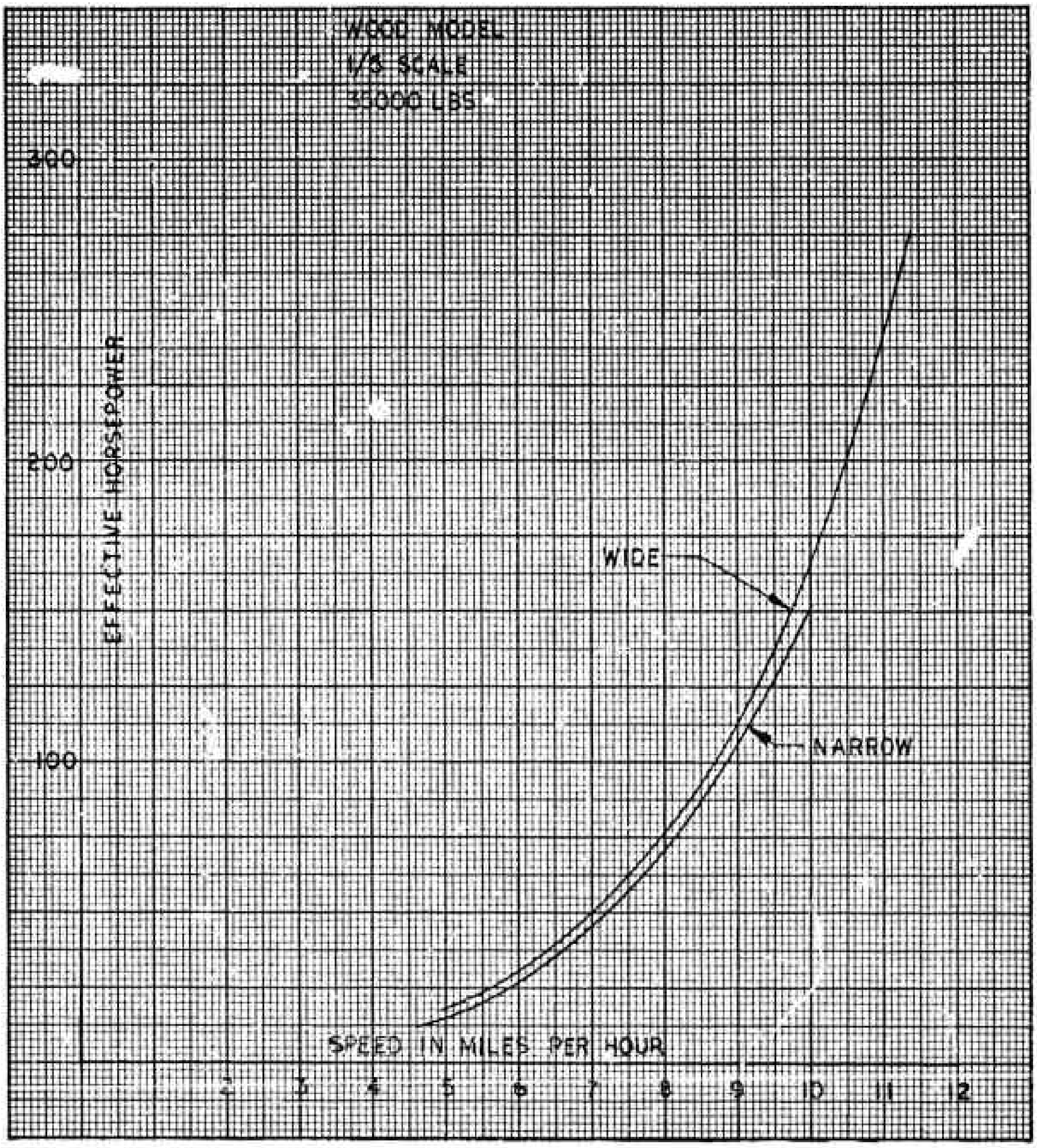


Figure 4-42 Round Bow (No. 2) and Flat Bow (No. 1)


WOOD MODEL
1/3 SCALE
35000 LBS
EFFECTIVE HORSEPOWER
300
200
100
WIDE
NARROW
SPEED IN MILES PER HOUR
2
3
4
5
6
7
8
9
10
11
12

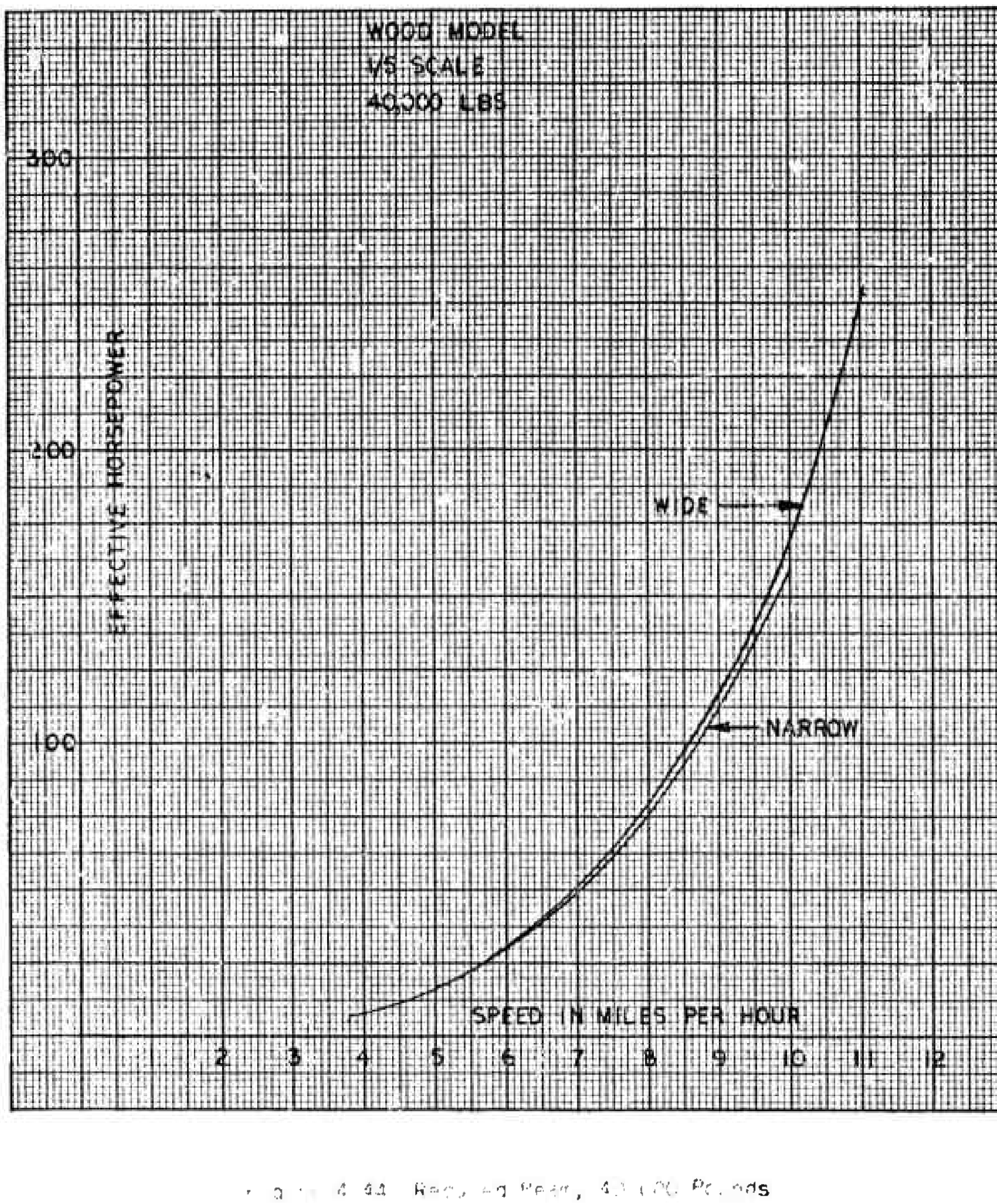


Figure 4.44 Required Power, 40,000 Pounds

wood, lacking the details of water grousers. Figure 4-16 shows that decreasing the beam of the model with real tracks would probably not produce a decrease in resistance as great as decreasing the beam of the wooden model.

4.2.11 Side Skirts. The benefit of side skirts -- coverings over the undercarriage and the return tracks -- has been firmly established by many experiments on tracked amphibians heretofore. Additional evidence is placed here for the record, to substantiate the findings of previous research. A comparison of the vehicle with and without side skirts is shown in Figure 4-45.

4.2.12 Trim. From the standpoint of towing resistance only, changes of static trim within reasonable limits produce only slight changes in resistance. Figure 4-46 shows an experiment in which static trim was changed through a range of 3.5 degrees from bow to stern while the vehicle was propelled by track at a constant speed. Stern trim slows the model. This was with long contravanes.

In self-propelled tests by tracks or by propeller the initial static trim does have some effect on required horsepower. The chief reason for this is that the craft changes its trim more when under way by its own power than it does when being towed. In Paragraph 4.2.7 it was remarked that the model trims down by the stern 2 degrees when equipped with stern baffles and very short contravanes. Consequently, if the initial static trim is by the bow, the trim when under way is just about optimum. Figure 4-47 shows the effect of varying trim with shortened contravanes. The angle of the contravanes can be adjusted, however, to produce very little change of horsepower with trim. Figure 4-48 shows the effect in an experiment with propellers.

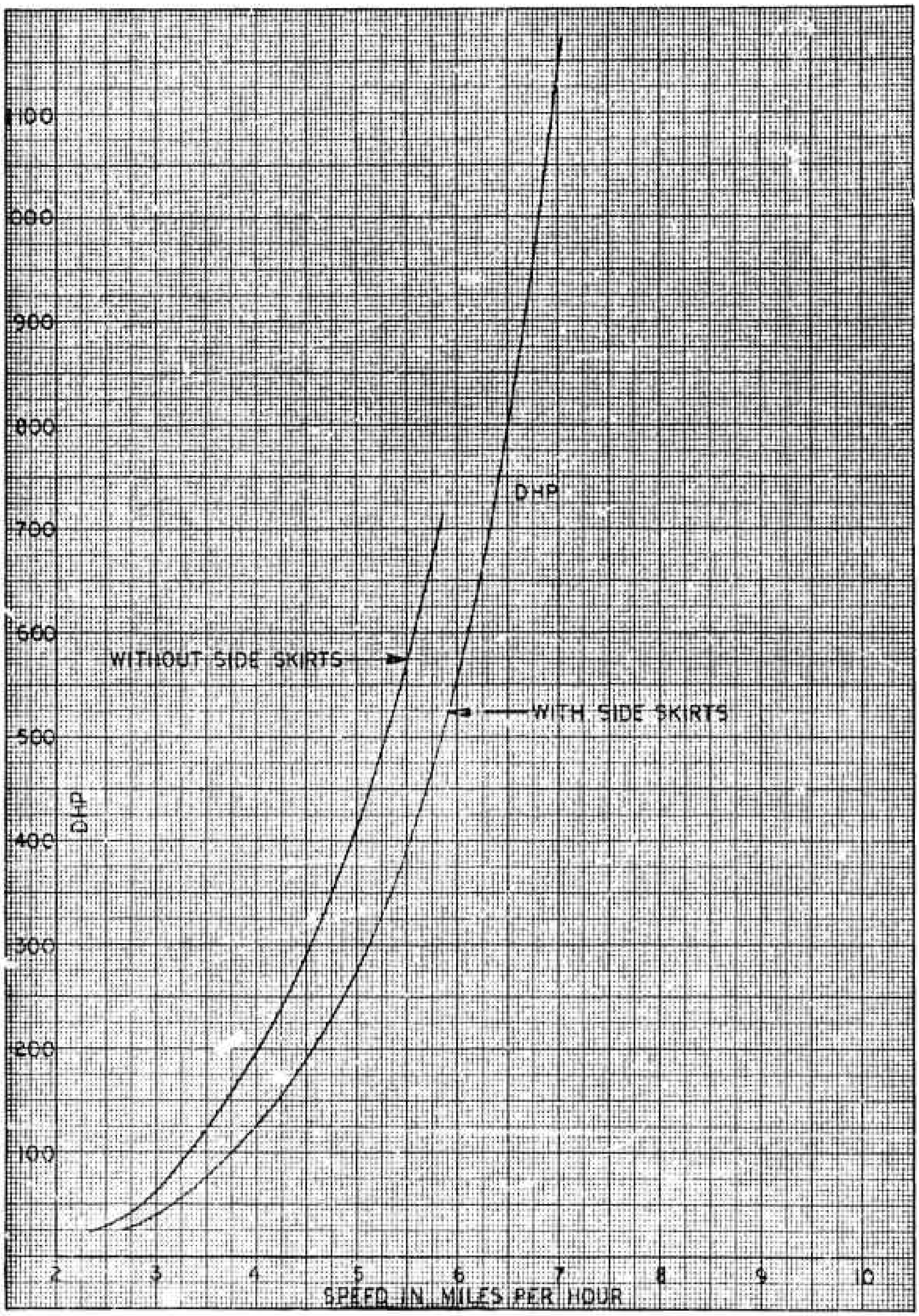
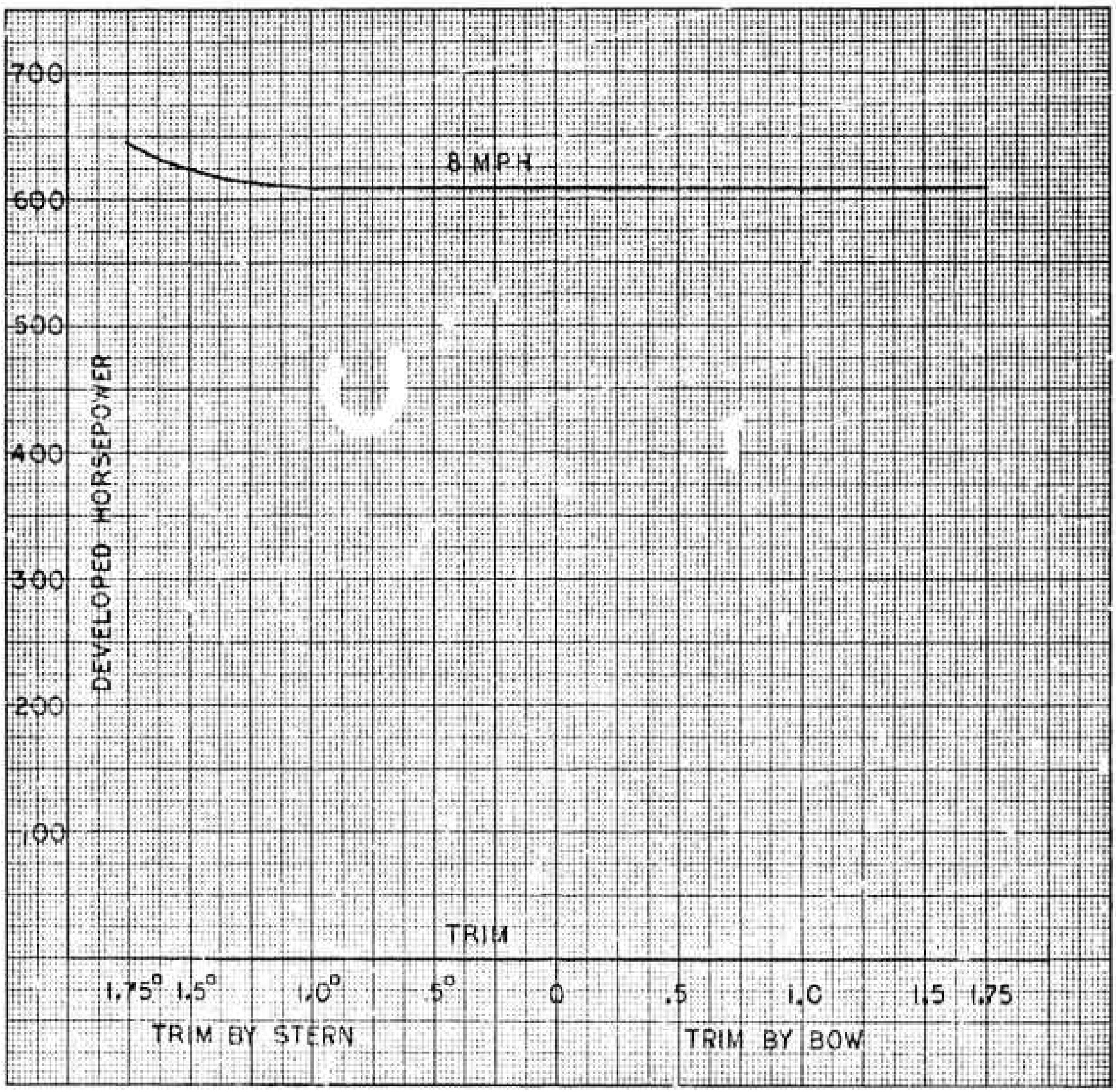
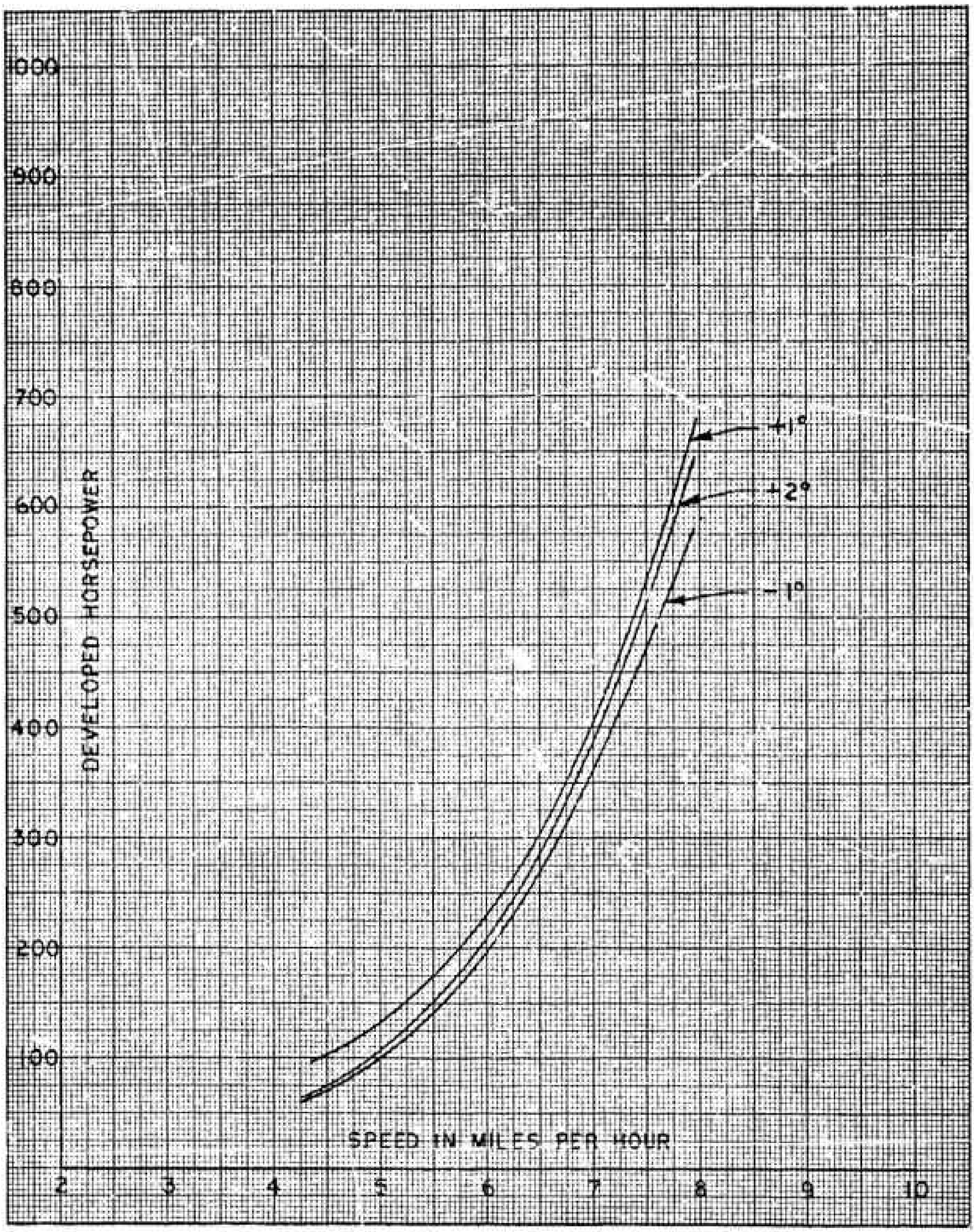


Figure 4-45. Side Skirts


700
600
500
400
300
200
100
DEVELOPED HORSEPOWER
8 MPH
TRIM
1.75° 1.5° 1.0° .5° 0 .5 1.0 1.5 1.75
TRIM BY STERN
TRIM BY BOW


1000
900
800
700
600
500
400
300
200
100
DEVELOPED HORSEPOWER
+1°
+2°
-1°
SPEED IN MILES PER HOUR
2
3
4
5
6
7
8
9
10

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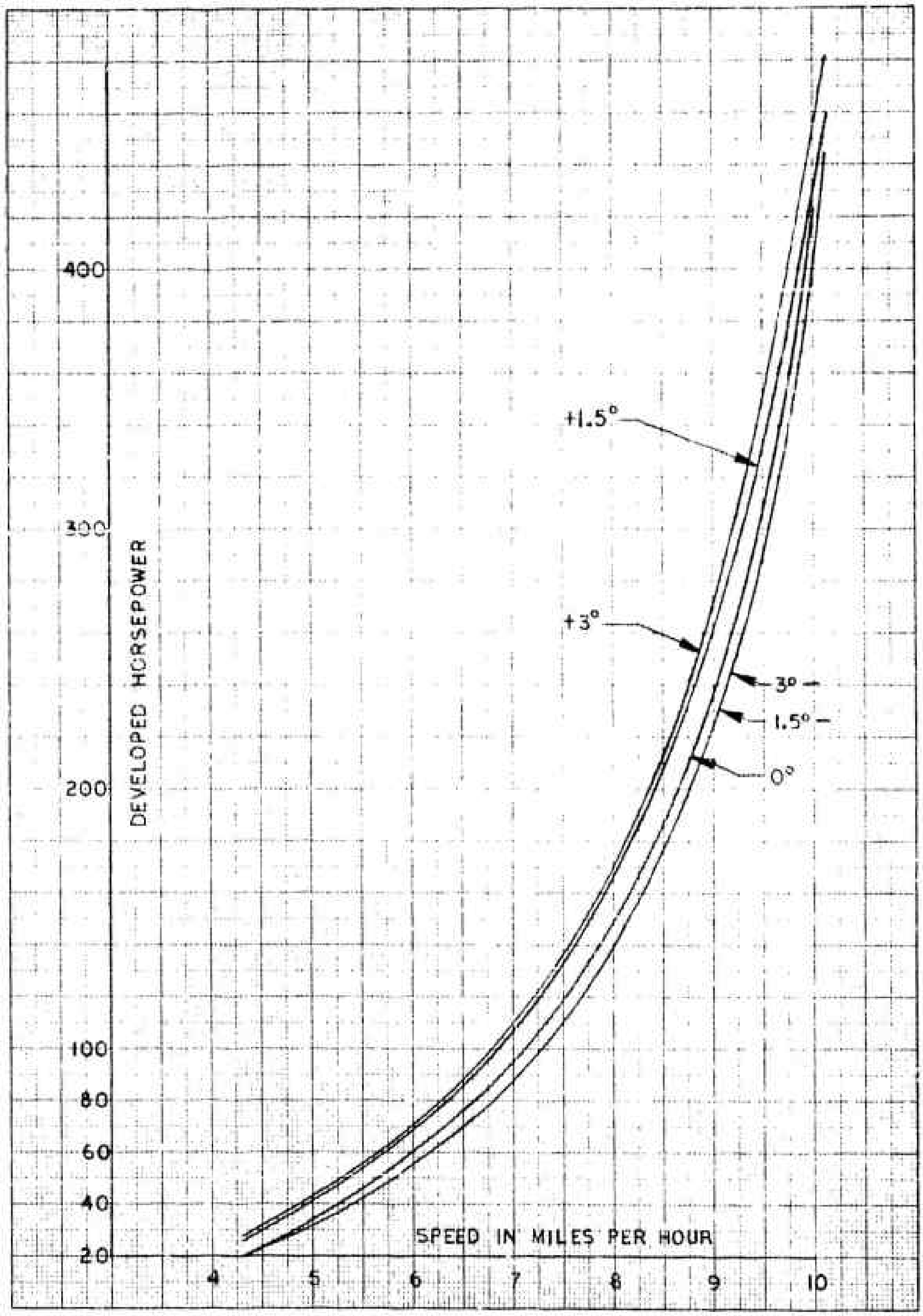

Figure 4-48 DHP by Propellers with Various Trims

It has been stated in the past that static bow trim resulted in better propulsive efficiency by tracks, and it has been taken for granted that the tracks had a "better bite on solid water" or some such reasoning. Observations during this testing program, however, are that initial static bow trim improves performance because the trim when under way becomes more favorable. Comparison of the track tests and the propeller tests substantiates this reasoning.

4.2.13 Weight. Heretofore it has been stated that the LVT performed as well or better when heavily loaded as when light. Previous vehicles have been slow craft when compared with the LVTPX12. Figure 4-49 shows that at low Froude numbers the required power of the track-propelled vehicle increases only a little with increased weight, but that at high Froude numbers the response to weight is much more marked. The same pattern is shown in Figure 4-50, made from towing tests of the wooden model. It may be true that some improvement in propulsive efficiency results from deeper immersion of the tracks, but nevertheless at high speeds the required power climbs more rapidly than the efficiency.

4.2.14 Backing Speed. The LVTPX12 will have no difficulty achieving adequate backing speed. Although in the test shown by the curve on Figure 4-51 the model was propelled backwards to only 5 MPH, the vehicle actually could go a little faster. Although the torque was very low in this experiment, the test had to be stopped at 5 MPH because the water was splashing over the stern of the open model.

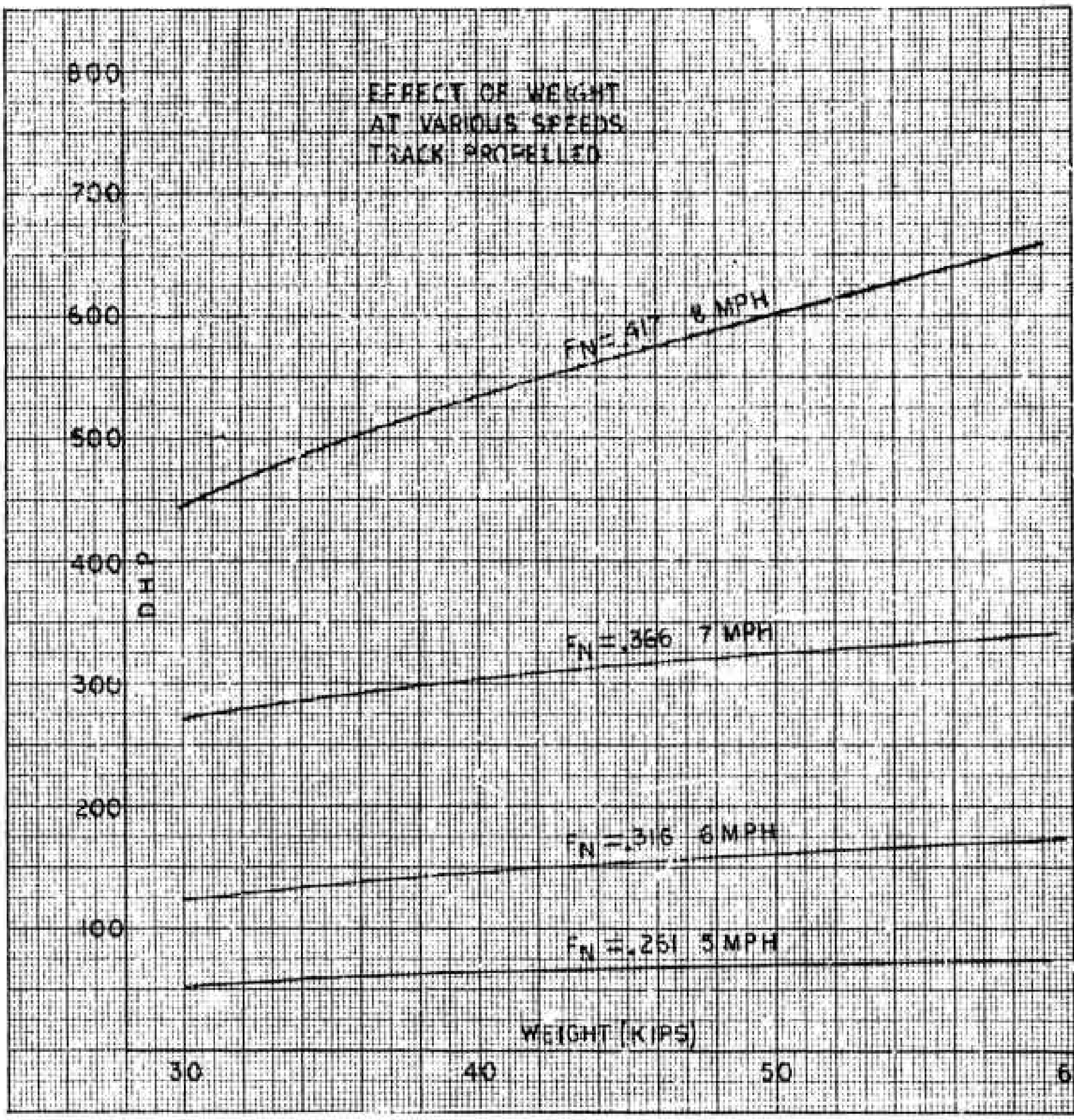

Figure 4-49 Weight-Speed-Power Relationship, Track Propulsion

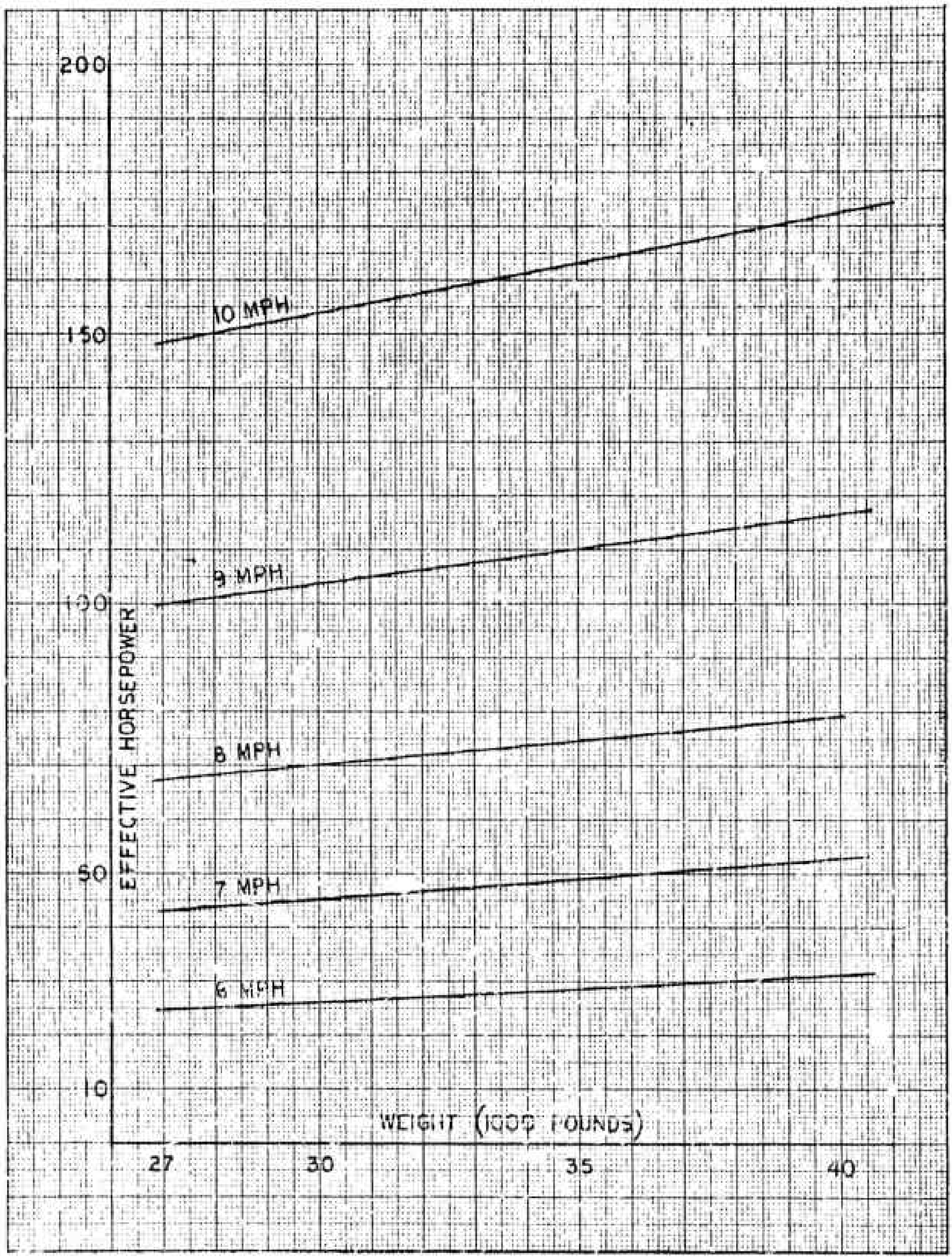

Figure 4-50 Weight-Speed-EHP, Wood Model

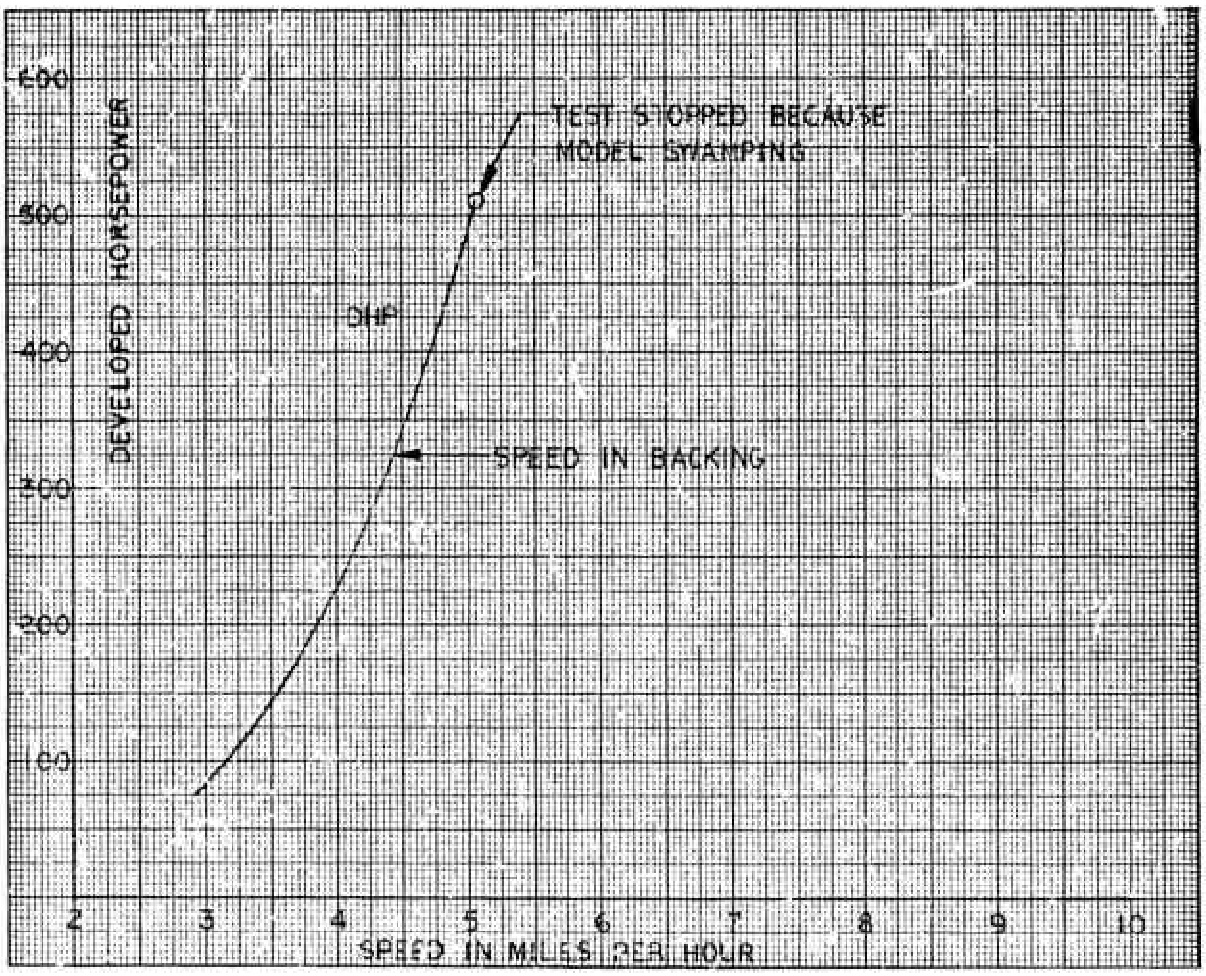

Figure 4-51 Backing Speed

4.2.15 Supplemental Thrust with Track Propulsion. At 8 MPH, the tow rope horsepower of the LVTPX12 weighing 50,000 pounds is only 90. Although the 15 percent propulsive efficiency of the track propelled vehicle at this speed is high for track propulsion, it is still not high when compared with propellers. On the possibility that it would pay to put a part of available engine power into the tracks and the balance into another device, a series of experiments was performed in which the model was towed at a series of fixed speeds while the tracks were turned. Thus for any given speed, a curve could be drawn showing the required supplemental thrust at that speed as a function of the horsepower put to the tracks.

The possibility that such a combination would pay seems worth considering because of the observable rapid decrease in required thrust as the tracks begin turning, this being particularly rapid as the tracks are sped up from zero speed to the water speed of the vehicle, as shown on Figure 4-52. The solution is obtained as follows: (1) For a given speed, find the required DHP to drive the vehicle with a given propulsive device. Translate this into the required SHP at the engine. Consider this the total available horsepower for a combination of track and auxiliary propulsion. (2) Find the DHP required to turn the tracks at some chosen slip ratio, say zero slip, and translate this into SHP by adding the frictional losses in tracks and gear train. (3) Subtract the SHP thus obtained, SHP_T, and SHP_P, and find the shaft horsepower remaining, available now for putting into the auxiliary propulsive device. This last SHP may be converted into thrust, as shown in Appendix Section 13.0. If the thrust thus resulting is greater than that required, then the power put into it has been profitably allocated, but if the resulting thrust is lower than that required to maintain the speed, the boat will slow down. In

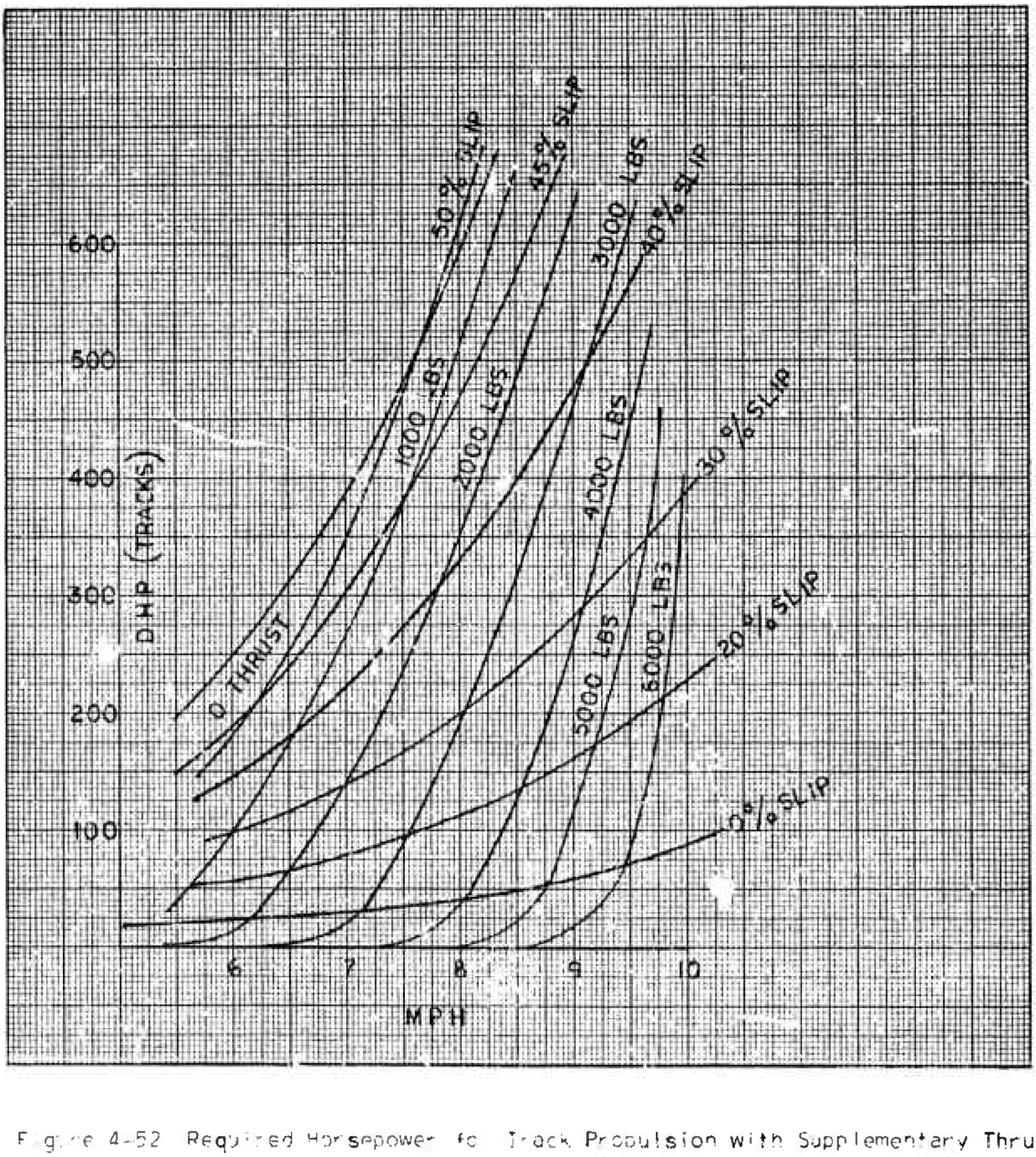

Figure 4-52 Required Horsepower for Track Propulsion with Supplementary Thrust

Appendix A, Section 13.0, the example chosen is at 9.55 MPH, as near to 10 MPH as the experimental results will go. Assuming the tracks are turning at zero slip, the required supplemental thrust is 5900 pounds, but the thrust obtainable from a pair of propellers, with an assumed propulsive efficiency of 44 percent, is only 5100 pounds at the remaining available power. Therefore, it would not pay to use part of the shaft horsepower to turn the tracks at zero slip.

4.2.16 Slip The ratio of the difference between track speed and speed of advance to track speed, called the slip ratio, is important chiefly for the purpose of designing the proper gear ratio for desired vehicle speed. The slip ratio is associated with propulsive efficienty, but because of frictional losses the relationship is not direct. In the tests showing optimum configurations, the slip ratio is close enough to 50 percent to use that figure in the design of gear ratio.

4.2.17 Articulating Grousers. As shown in Paragraphs 4.7.5 and 4.7.15, a large proportion of power is lost in churning the water by the return tracks. While this loss cannot be entirely eliminated, because of the roughness of the track pads themselves, it is evident from the low resistance shown in Figure 4-40 that the grousers do contribute a large share to the resistance. (See also Figure 4-16, where the wooden model had only dummy tracks.) The power lost by the return grousers is illustrated proportionally to the resistance of the bottom grousers shown by the towing tests of Figure 4-40. One way to diminish this loss would be to invent a feathering bucket, which would exert maximum force while going rearward and offer minimum resistance on the forward trip, like a duck's foot. Of several possible designs for

such a feathering bucket, one is illustrated by sketch in Section 9.5, where its operation and its objections are explained. In addition to the predictable faults of this track for land propulsion, the extent of benefits in water are unpredictable, so that experiments with a model of it are not very likely to repay their cost.

An estimate of the possible benefit of feathering buckets may be obtained from Figure 4-52. At zero slip, the power required to turn the tracks at 8 MPH, with no frictional losses, is 42 horses. Assume that all of this power was expended in churning the water with the return track. If the track had no grousers at all, it still would have resistance because of its links and lugs. If the power loss were reduced by 50 percent -- a liberal allowance -- the reduction in required power would be only 21 horses, or 3.5 percent of 600 at 8 MPH. This reward is not great enough to offset the problems examined in Section 9.0.

4.2.18 <u>Optimum Tracked Configurations</u>. From the large number of tests in this program, it is clear that by adopting the best features of the model a water speed of 8 MPH in calm water can be attained. It is also clear from the curves, however, that to press for much more than 8 MPH by track propulsion will call for great increases in power. Bow fenders and stern baffles are found to be indispensible. The particular features that must be incorporated in the prototype, for 8 MPH or better and the general conditions for operation, are as follows:

- Bow fenders wrapping 150 degrees around the front sprockets must be adopted. Since for land operation these fenders must be retracted, the means for retracting them must be provided in the design and must be accepted by the operators.
- Stern baffles must close the opening behind the rear sprocket rather tightly. These baffles should have a lip, or contravane, extending horizontally rearward, about 4 inches. Although longer contravanes, capable of being adjusted both in angle and elevation, will give better control over dynamic trim, they generally will be objectionable in operation. These stern baffles must also be retractable for land operation.
- The bow of the craft must be as fine as allowable by the demands of land operation and necessary displacement. The boat bow, #3, used on the model is about as fine as can be designed within practical limits. This bow has a developable surface, so that furnacing and die-shaping of plates will not be necessary in fabrication. Elimination of the chines would produce some improvement in performance, but the increased cost of fabrication would be excessive.
- The stern may be shaped according to the requirements of the geometry and center of gravity of the stern ramp and to whatever extent possible for the looks of the vehicle in profile. Since it is impossible to shape the stern enough to decrease resistance, the hydrodynamic aspects of the stern may as well be ignored.

- Side skirts over the undercarriage are necessary. They need have no great strength. It makes no difference hydrodynamically what they are made of, as long as they are smooth. They should extend downward to the level of the hull bottom.
- The best grouser of the four tested was #1. This is an excellent grouser, but it could possibly be a little better. Although the amount of possible improvement in the grouser is not known, it is reasonable to expect that the improvement could not be very large.
- If the installed horsepower is in excess of the minimum necessary to supply 600 DHP through the tracks, the gearing should be designed to produce slightly more than 16 MPH track speed at full engine speed. Gearing to produce 16 MPH track speed at 97 percent or as much as 96 percent of rated engine speed would provide an adequate design margin.

Figure 4-53 is a set of lines drawings in which the foregoing stipulations are incorporated.

4.2.19 Prediction of Prototype Performance. The statement in Reference (16), that the coefficient for extrapolation of model data is some variable function of speed, has led to the belief by some people that predictions from self-propelled model tests should be made by some multiplier greater than scale factor to the power of 3.5. That statement was not documented, however, and it is not supported by scientific analysts. The only comparison between

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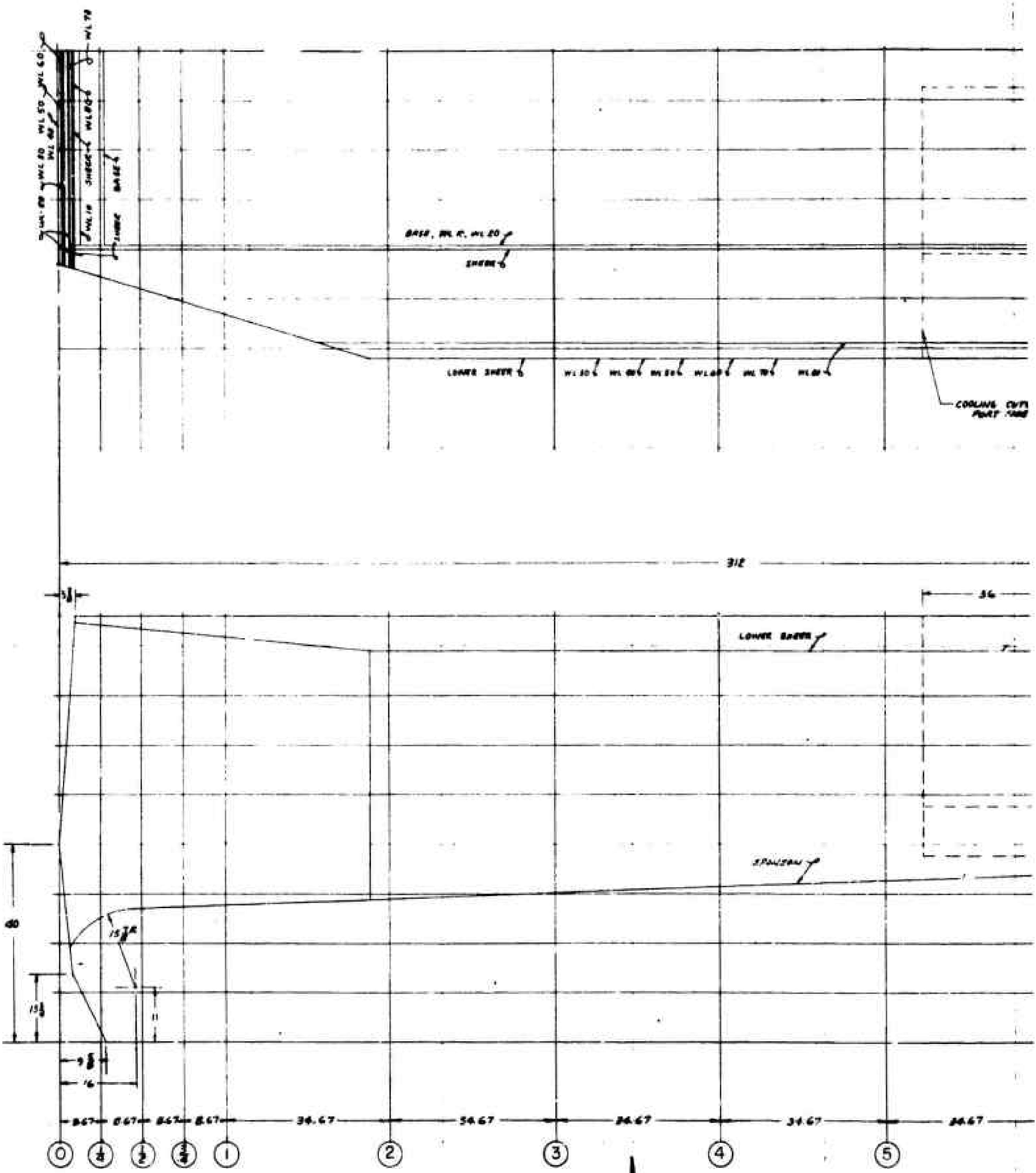
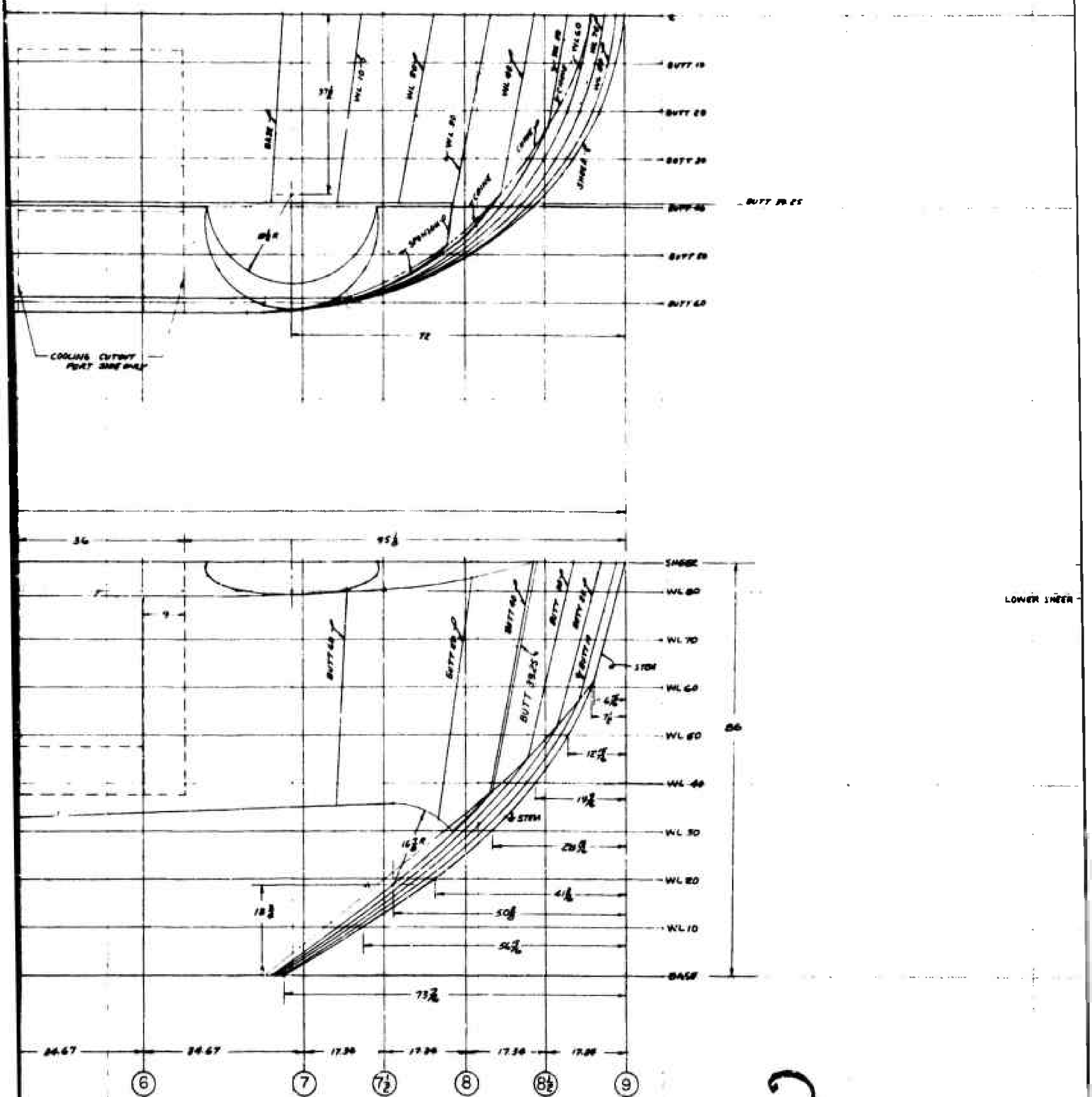
CORPORATION

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COOLING CUTOUT
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36
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LOWER SHEER
WL 80
WL 70
WL 60
WL 50
WL 40
WL 30
WL 20
WL 10
BASE
STEM
86
24.67
17.38
6
7
7½
8
8½
9

2

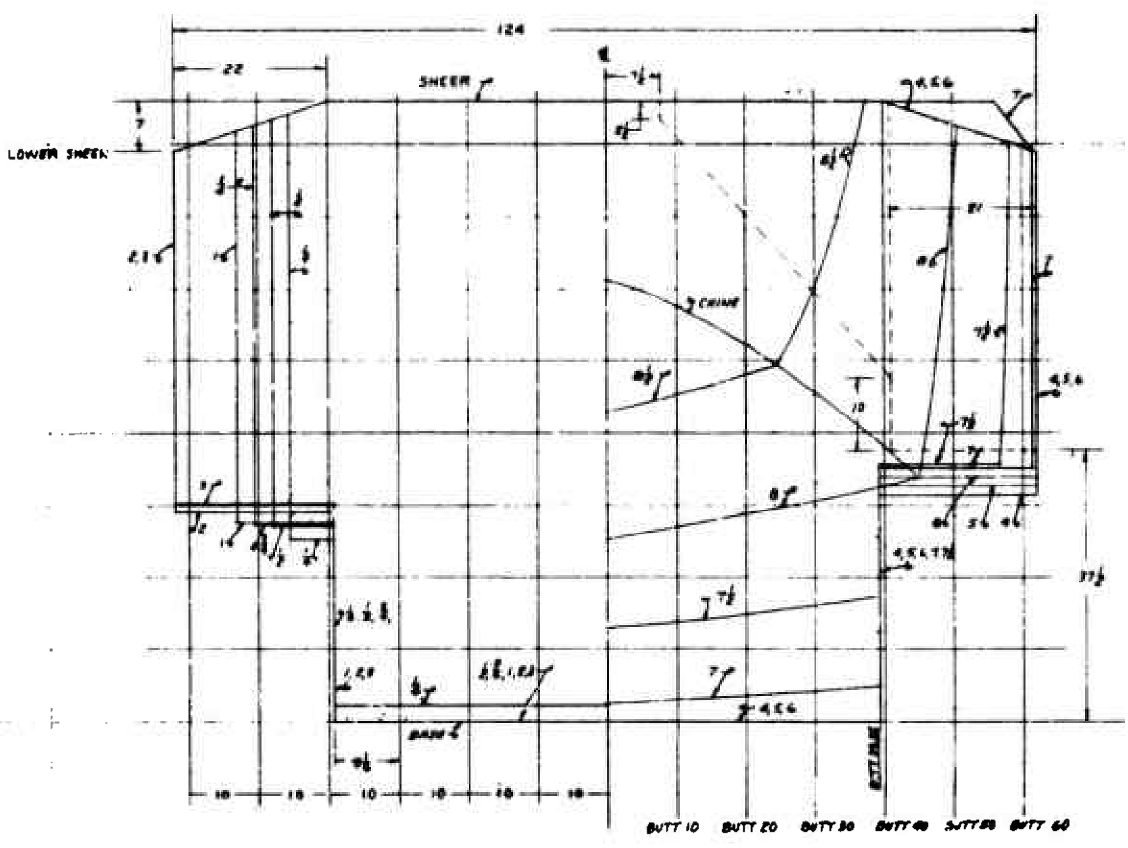


FIGURE 4-53 LINES OF TRACK PROPELLED LVTPX12

model and prototype performance is analyzed in Appendix A, Sections 10.0 and 11.0. This analysis shows that the power of the prototype can be predicted very closely by use of scale factor to the power of 3.5. In the tests of the full scale LVTP5 by the U.S.N.E.E.S. the steering losses were not accounted for. These could have produced the slight difference between predicted power and power actually required. As for towing resistance, the wash of the tug's propeller and the wake of the towing cable make the results of the N.E.E.S. towing experiment subject to question. The best evidence shows that for prediction of delivered horsepower the use of scale factor to the power of 3.5 is correct.

Therefore on the basis of this evidence and with the support of rational analysis, (Appendix A, Section 11.0) Figure 4-54 shows the required shaft horsepower for the LVTPX12 with boat bow, track propelled. This curve was obtained by adding predicted frictional losses in the track system (Section 9.5) plus the predicted losses in the power train (Section 8.0) to the DHP curve from the test with stern baffles and short contravanes. The justification and evaluation of these frictional losses are contained in the referenced sections, and these evaluations should be considered in interpreting the reliability of Figure 4-54.

4.3 Model Tests With Auxiliary Power. For the original proposal on the design of the LVTPX12, tests were performed by the Ship Hydrodynamics Laboratory of the University of Michigan with a 1/5 scale model propelled by a single Kort nozzle propeller of 27 inch diameter. (Figure 4-12 and 4-13). Because of the usefulness of these data in the subsequent work, the curves of these tests are properly a part of this report. The single propeller, although demonstrating

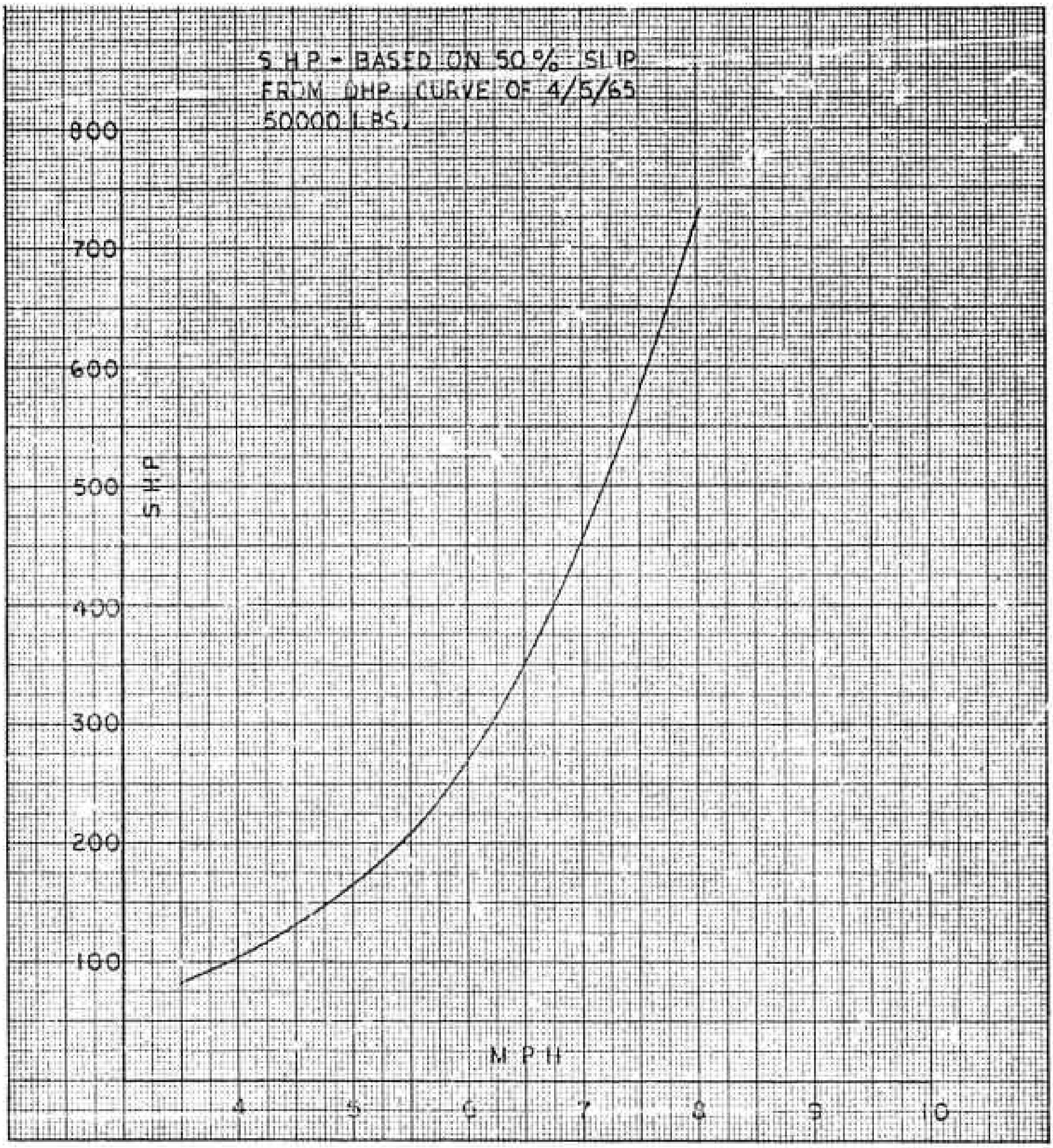


Figure 4-54 SHP at Engine, Track Propulsion

that the vehicle could be pushed at 10 MPH with reasonable power, was subject to certain objections:

- The ground clearance of the vehicle being only 16 inches, the propeller had either to be in a vulnerable position or else to be so high that the hull in front of it produced a high thrust deduction and low hull efficiency.
- Although a design for housing the propeller during land operation was provided, the propeller could not be used in the housed position.
- If the propeller were operated when partially raised, a high trimming moment would be applied to the vehicle, and the horizontal component of thrust would be very small.
- Being essentially like an outboard motor, the single propeller would be an excellent steering device, but the steering mechanism cannot be said to be mechanically simple, even though successful installations of this nature are already being made. A heeling moment would result during a quick turn, and could capsize the craft in certain sea conditions.

A pair of side propellers solves the vulnerability problem. While these would be subject to damage in case the vehicle sideswiped something with the propellers outboard, they would be well clear of the ground upon landing. Although they would not get the benefit of a high wake, neither would they be penalized by high thrust deduction. The side propellers can be operated in any position without placing a trimming or heeling moment on the vehicle.

Although they will not produce high thrust in the housed position, they nevertheless will produce enough to propel the vehicle. As for speed, if propellers are damaged, the tracks will still provide better than 6 MPH, even without the bow fenders.

The only objection, in fact, to the twin side propellers from an operational standpoint would be the steering problem. This, however, can be handled by the use of controllable pitch propellers. (See Section 4.9). An additional attraction of controllable pitch is the freedom to select the best pitch and RPM for any speed and loading.

4.3.1 Experiments with a Single Kort Nozzle Propeller. In interpreting the data from the single propeller on the wooden model and from the twin propellers on the metal model, the difference in resistance described in Paragraph 4.2.3 should be kept in mind. In Figure 4-55, the towrope horsepower (EHP) of the wooden model was obtained by extrapolating the towrope resistance of the wood model to 50,000 pounds. The metal model, on a scale of 1/4.5, was equipped with the boat bow (bow #3) and bow fenders in this test. Beal's report that the resistance of a wooden model was 84 percent that of a metal tracked model (Reference 20) supports this evidence. By applying the percentage of increased resistance to the curve for the wood model, the prediction of the power required to be delivered at the single Kort nozzle propeller of 27 inch diameter may be approximated as in Figure 4-56.

Some cavitation was observed during tests on the 27 inch single propeller (See Proposal). These tests were all at 45,000 pounds. The increased load

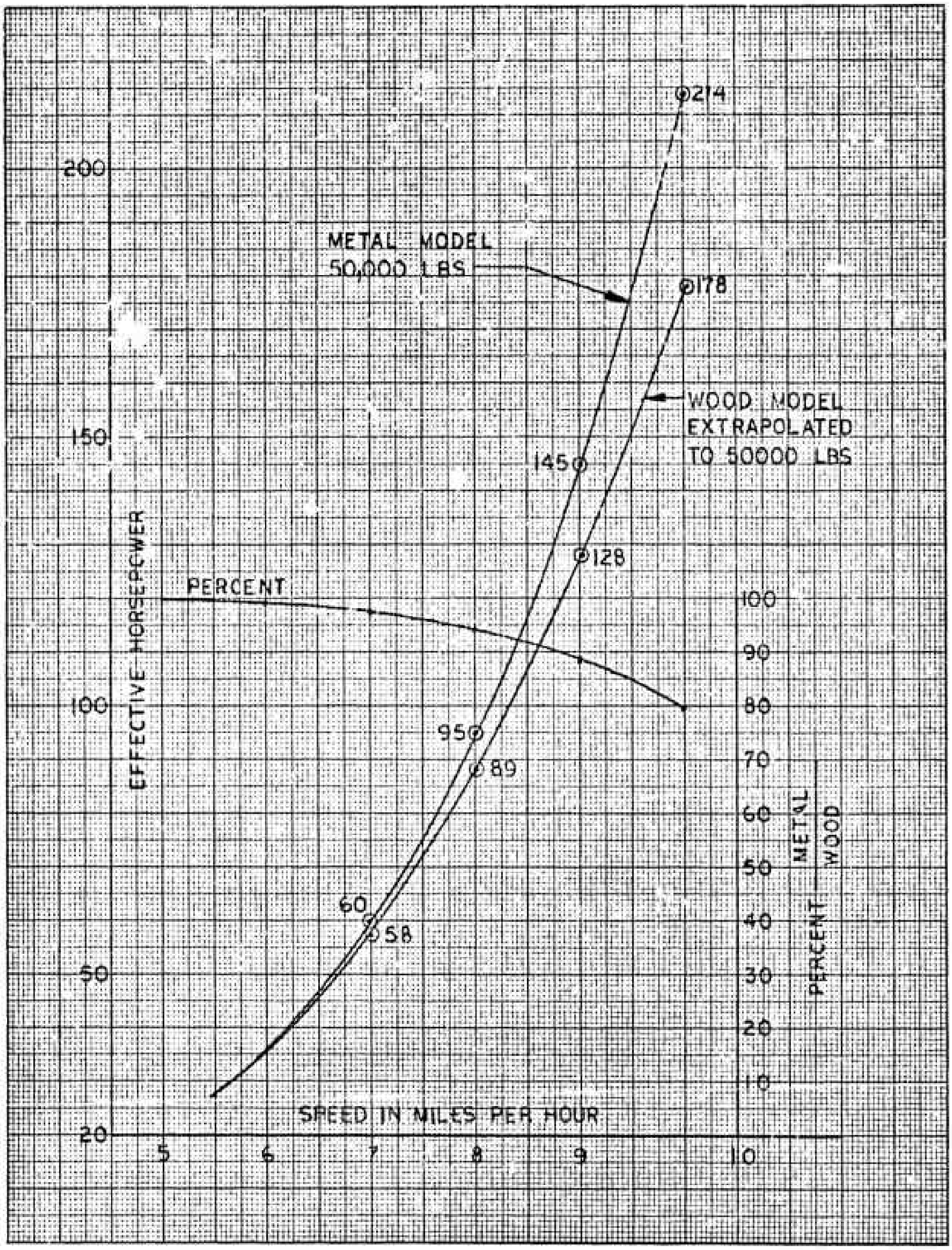


Figure 4-55 Towrope Horsepower Compared

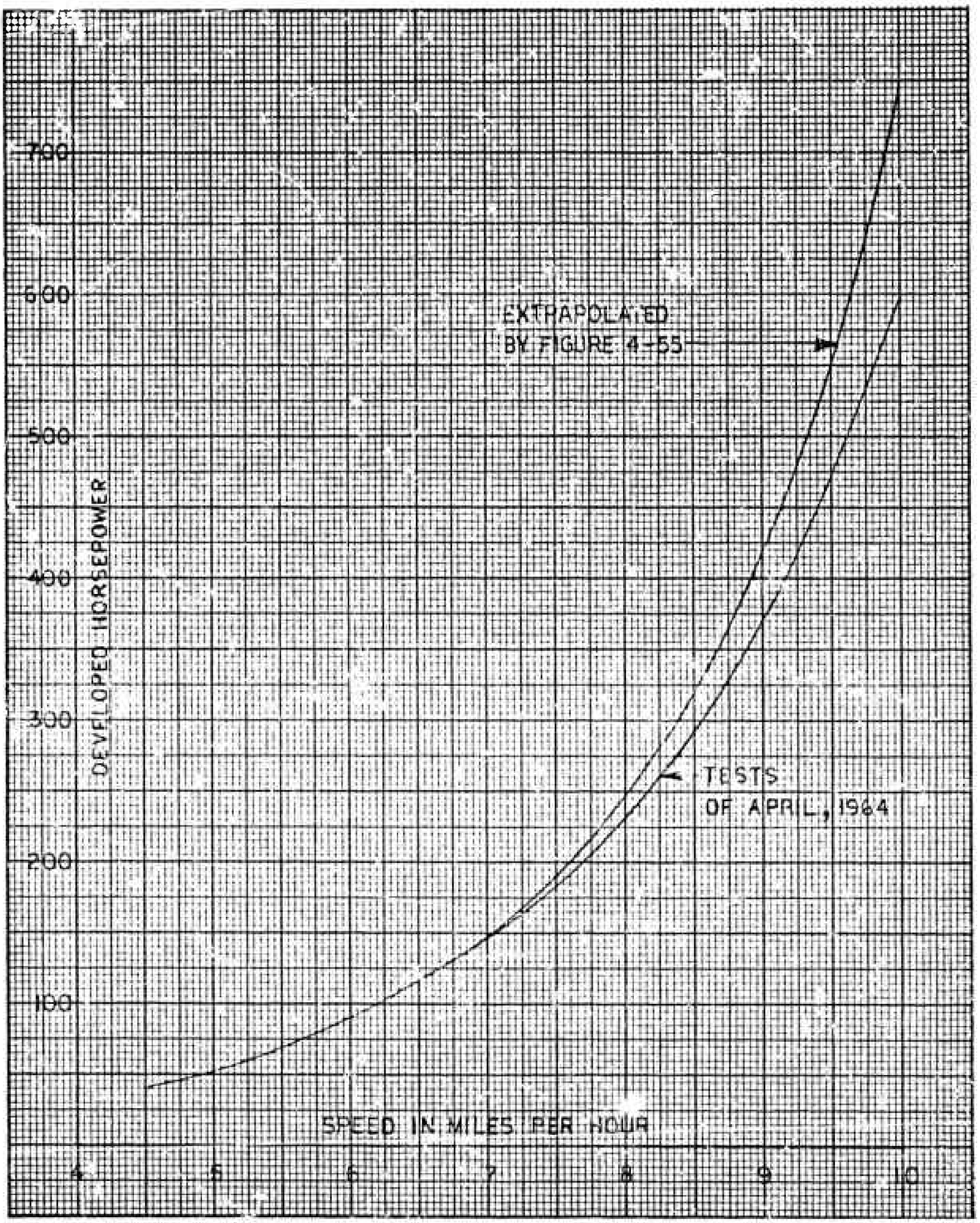


Figure 4-56 DHP With Single Propeller, Corrected

on the propeller that would have resulted with a displacement of 50,000 pounds, plus the increased resistance from using a model with metal tracks, would of course aggravate the cavitation. Inasmuch as the vehicle would be expected to operate in water only 20 percent of the time, however, the cavitation would produce no serious damage to the propeller blades, even when quite pronounced.

4.3.2 Experiments with Twin Side Propellers. Because model propellers much less than 7 inches in diameter are subject to scale effects, the propellers used in the tests measured 6.46 inches. This amounts to 29 inches on the prototype, whereas the space available over the track sponsons will admit an outside diameter, over the Kort nozzle, only 26 inches. Hence the views of the model in Figures 4-10 and 4-11, and the views of the propellers when housed in Figures 4-57 and 4-58, make the propellers appear slightly larger than they would be on the prototype. Lest there be some question on the use of a model propeller not corresponding to the prototype propeller, let it be understood that the purpose of a model self-propulsion test is to obtain the hull efficiency, and that it makes no difference what model propeller is used, so long as it is large enough to avoid propeller scale effects.

Several views of the model with twin propellers in the tank are shown in Figures 4-59 to 4-65. To simplify the driving system, flexible shafts were used. The wooden fairingblock attached to the shaft ahead of the propeller shows in Figure 4-59 where the model is at rest. The purpose of this block was to close the water behind the shaft, so that it came back solid before reaching the propeller. Although the resistance of the model would be

Figure 4-57 Propellers Housed - Side View

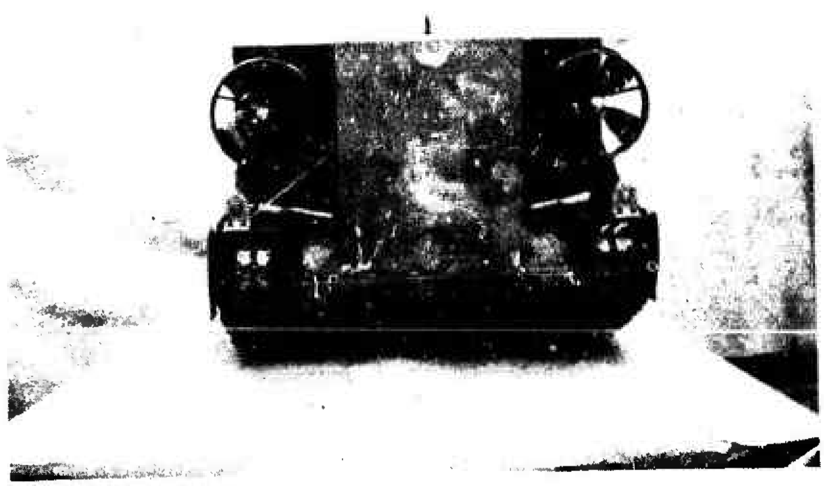


Figure 4-58 Propeller Housed - Stern View

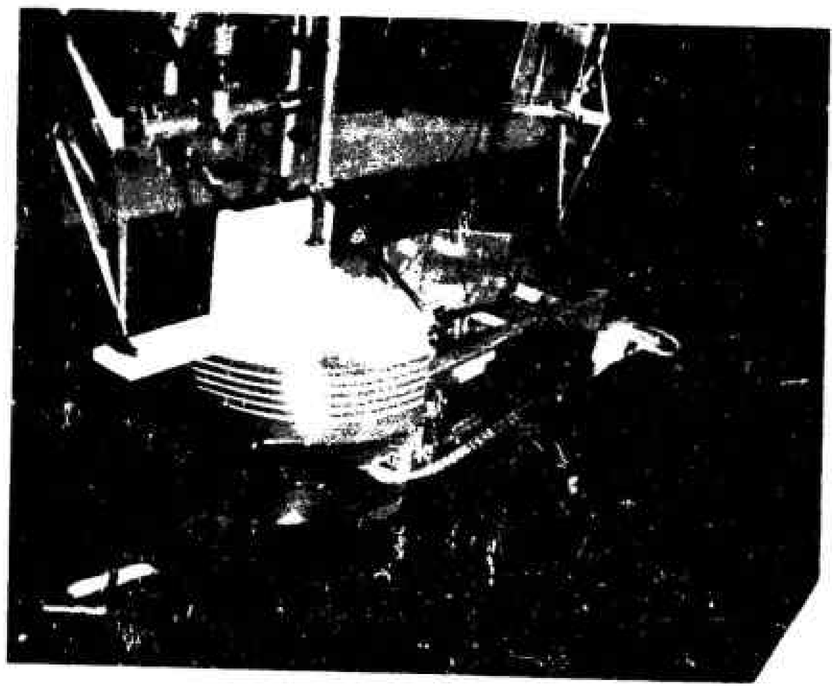


Figure 4-59 Model at Rest, Zero Trim, 50,000 Pounds

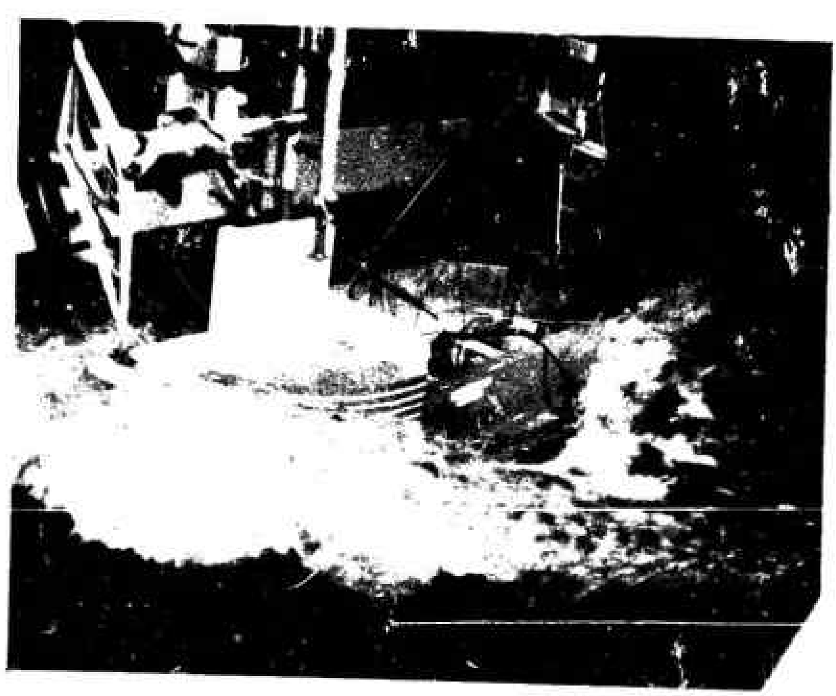


Figure 4-60 10 MPH, Zero Trim, 50,000 Pounds

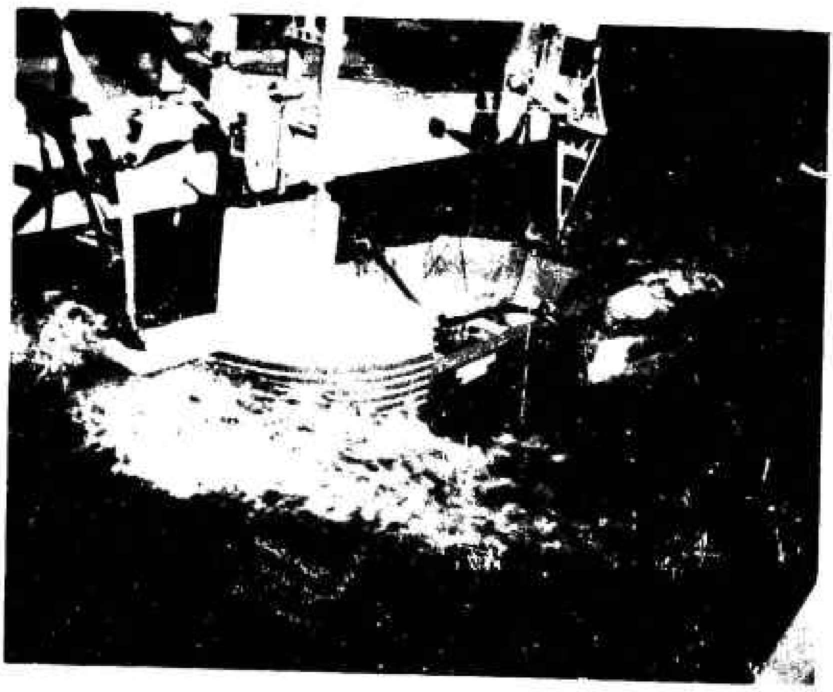


Figure 4-61 9 MPH, Zero Trim, 50,000 Pounds - Wave 4.5 Inches from Deck

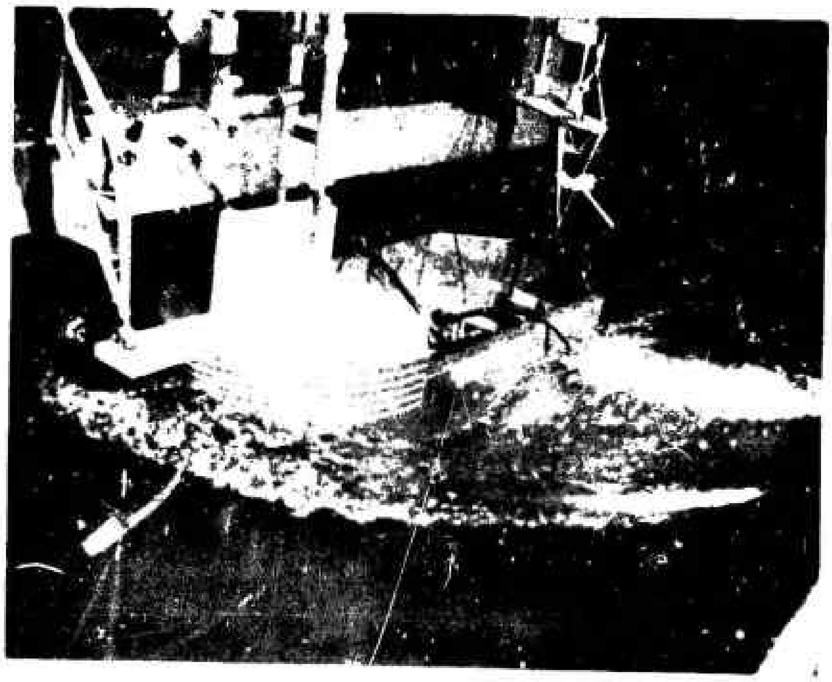


Figure 4-62 7.25 MPH, Zero Trim, 50,000 Pounds - Wave 16 Inches from Deck

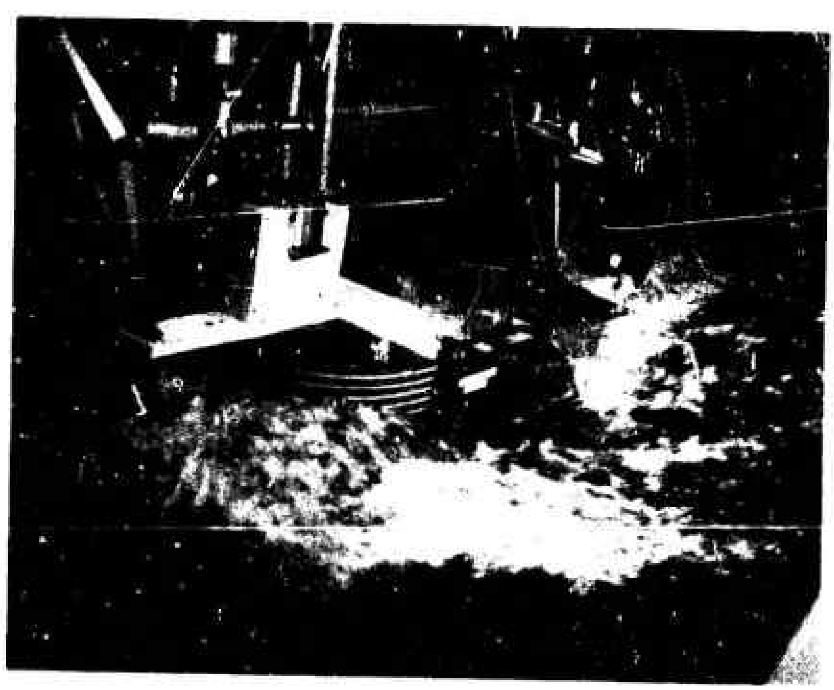


Figure 4-63 10 MPH, Trim 16 Inches Down by Stern, 53,000 Pounds - Wave 9 Inches from Deck

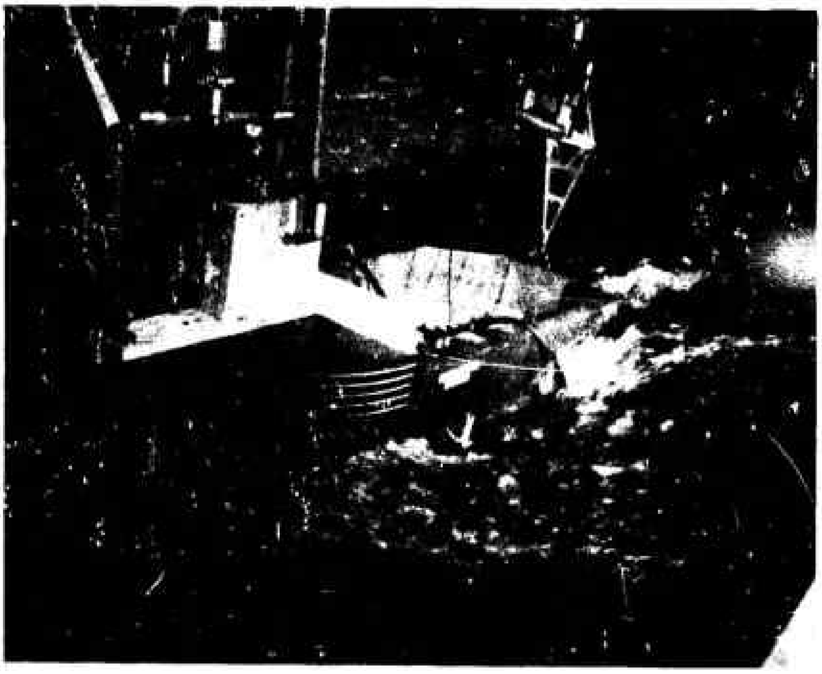


Figure 4-64 9 MPH, Trimmed 16 Inches by Stern, 53,000 Pounds - Wave 13 Inches from Deck

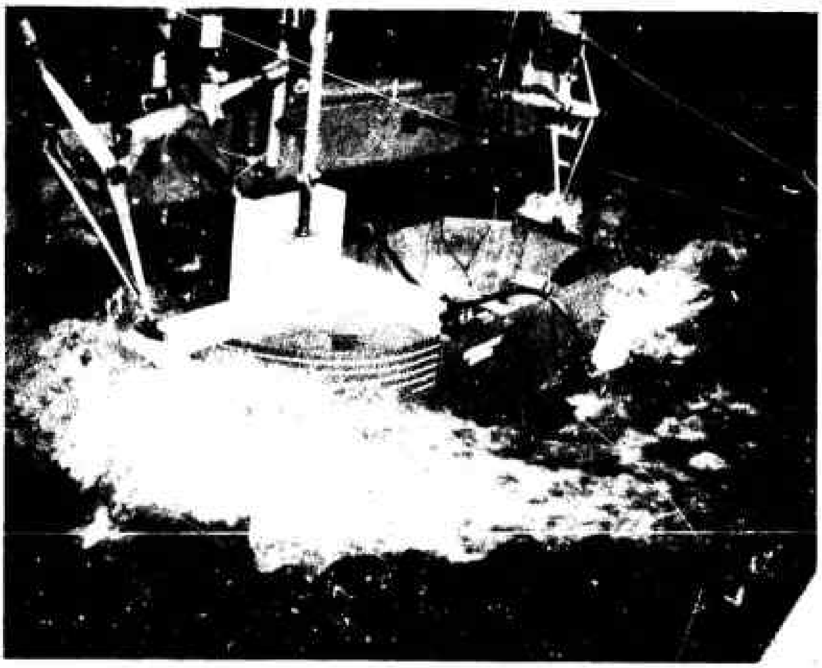


Figure 4-65 10 MPH, Trimmed 8 Inches by Bow, 43,000 Pounds - Wave 4 Inches from Deck

lower without the water grousers on the tracks, and hence the required propulsive power would be lower. (See Figure 4-66), the tests were performed with the water grousers on as shown in Figure 4-59. As is shown in Section 8.9.2, there is no advantage in selection of an engine of lesser power unless the weight can be reduced. Since the use of a smoother track would not reduce required power enough to adopt a lighter engine, the LVTPX12 might as well be able to go at a good speed with tracks or 10 MPH with propellers, at the option of the operator.

The model propeller was surrounded by a short Kort nozzle, of length-diameter ratio 0.27. The reason for adopting such a short nozzle was that the propeller was behind a drive pod of 10 inch diameter. The short nozzle provides protection for the wheel, but at the same time supplies a small amount of thrust while avoiding large reduction in the diameter of the wheel. Although the length-diameter ratio for a heavily loaded screw recommended by Van Manen(Reference 3 and 5, Appendix A) is usually .5, such a long nozzle would reduce the wheel diameter in this case to 22 inches under the limitation of 26 inches to outside diameter of nozzle.

The particulars, together with open water curves, of the model propellers are given in Appendix A, Section 14.0.

A wake test, with an independent propeller inside the nozzle but not attached to the model, provided the results shown in Figure 4-67. This chart shows the revolutions per second for the wake wheel in open water and in the nozzle alongside the model. The variations in wake with speed are caused chiefly by the movement of the side wave aft observable in

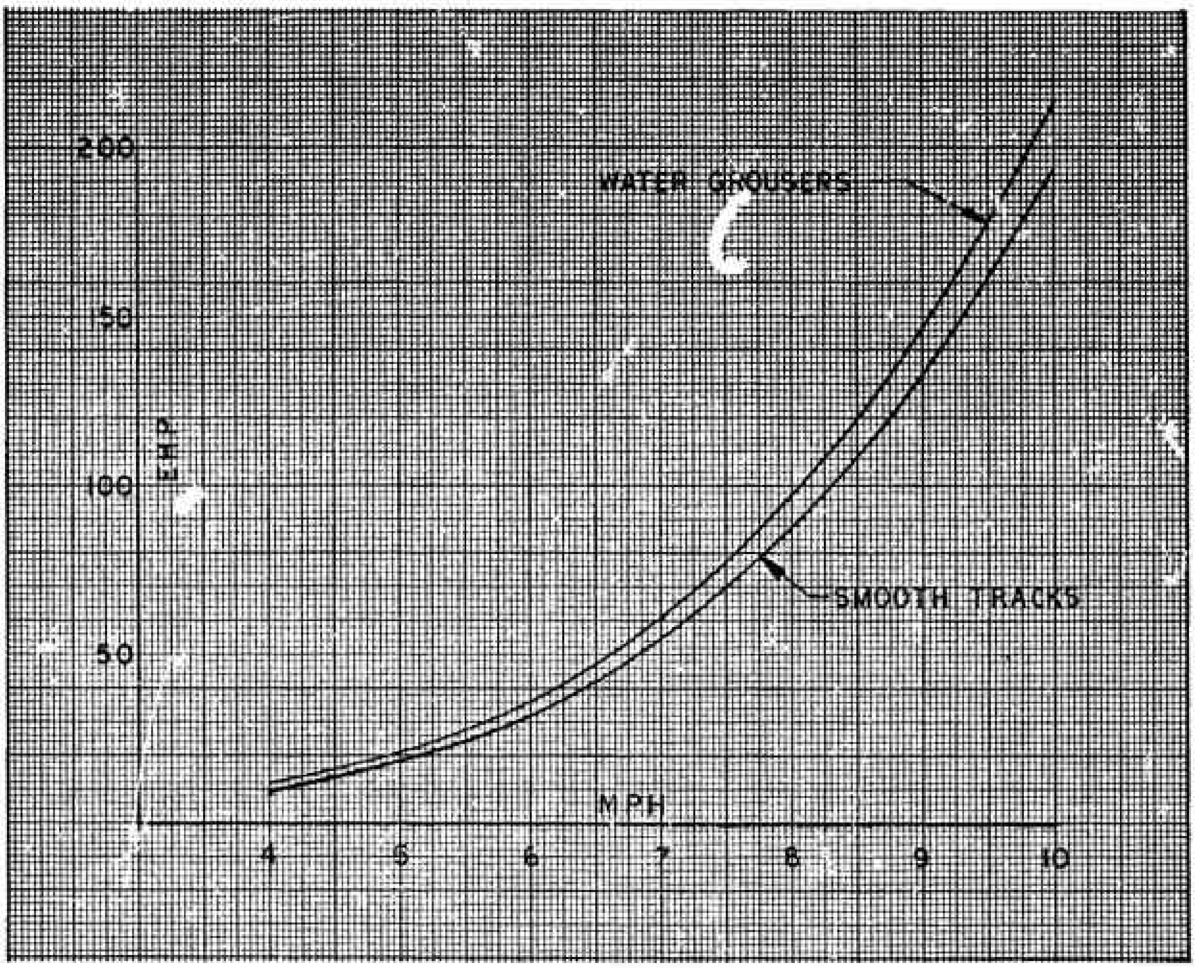
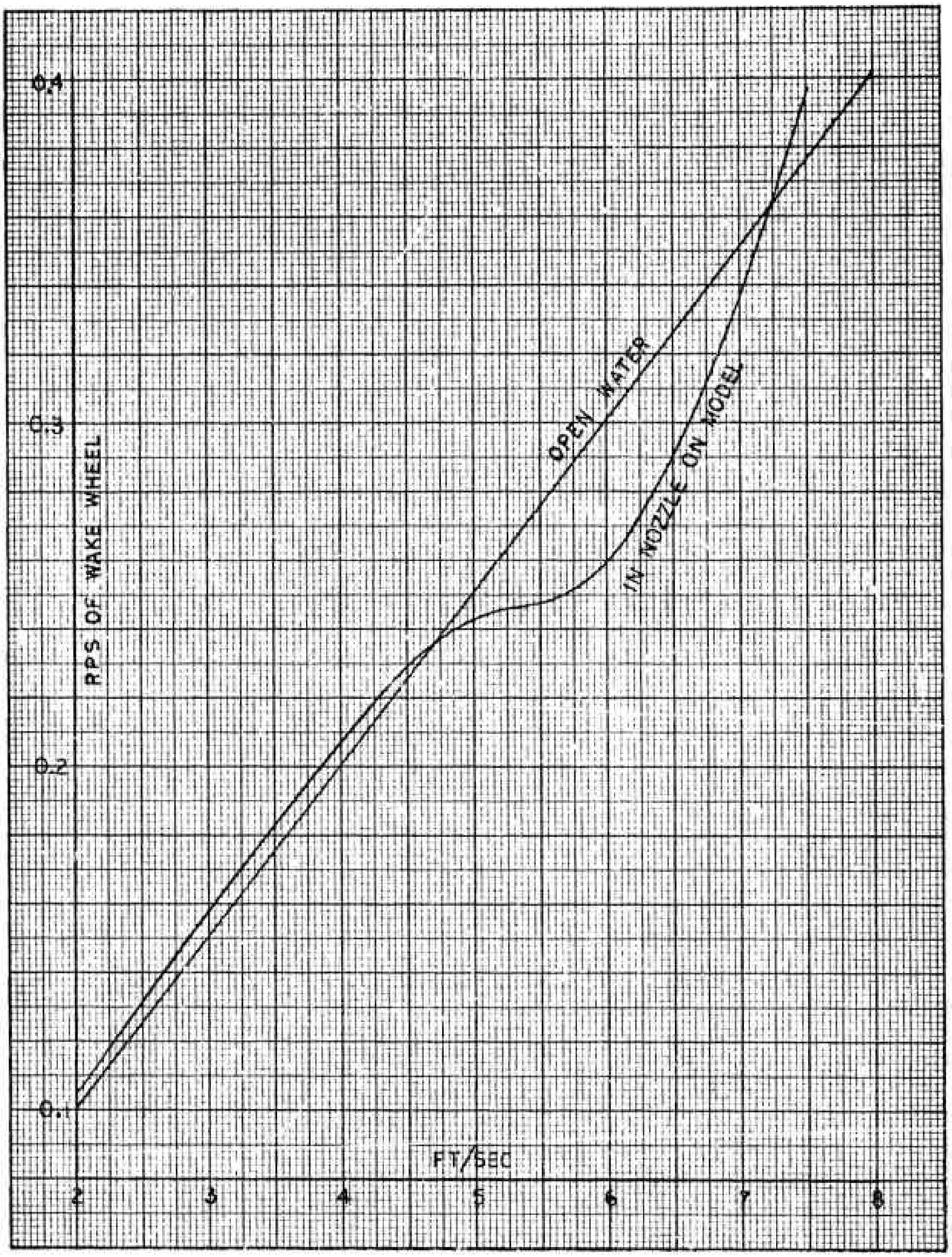


Figure 4-66 Comparison of Effective Horsepower -- Regular Water Grousers, and Grousers Completely Covered with Tape


0.4
0.3
0.2
0.1
RPS OF WAKE WHEEL
OPEN WATER
IN NOZZLE ON MODEL
FT/SEC
2
3
4
5
6
7
8

the photographs of the model at several speeds, Figures 4-60 to 4-65 At 10 MPH, the wake has a positive value of 5 percent.

During the tests, the propellers were observed closely for any cavitation visible to the eye. At 10 MPH a trail of hub cavitation was visible, but no blade cavitation could be observed. The inside of the nozzles were painted, and the paint stayed intact during the many tests, showing that no serious nozzle cavitation was present. The prototype wheels will be only 24 inches in diameter, however, and some cavitation, particularly nozzle cavitation, is to be expected. At the same time, these wheels will have 43 percent more disc area than the single propeller previously tested. That propeller did not suffer critical cavitation. Moreover, the projected area ratio of that propeller was only 50 percent, while the projected area ratio of the blades on the prototype will be 80 percent. For these reasons, the cavitation of the twin screws should be less than that of a single screw, since their net blade area is 130 percent more.

Since finding e_h is the first purpose of self-propelled tests, and since the propeller used in the tests is not a model of the prototype propeller, the curves of DHP as a function of speed obtained in the tests are useful only for purposes of propeller design. They are not curves of horsepower required in the prototype. To satisfy any interest, however, they may be inspected in Figure 4-66 These curves show both EHP and DHP. The propulsive efficiency at 10 MPH for these model propellers was 51 percent. Although these propellers scaled 29 inches in diameter, they were not the ideal design for this installation, either in pitch, blade area ratio, or blade section. While the efficiency would be reduced by the change in

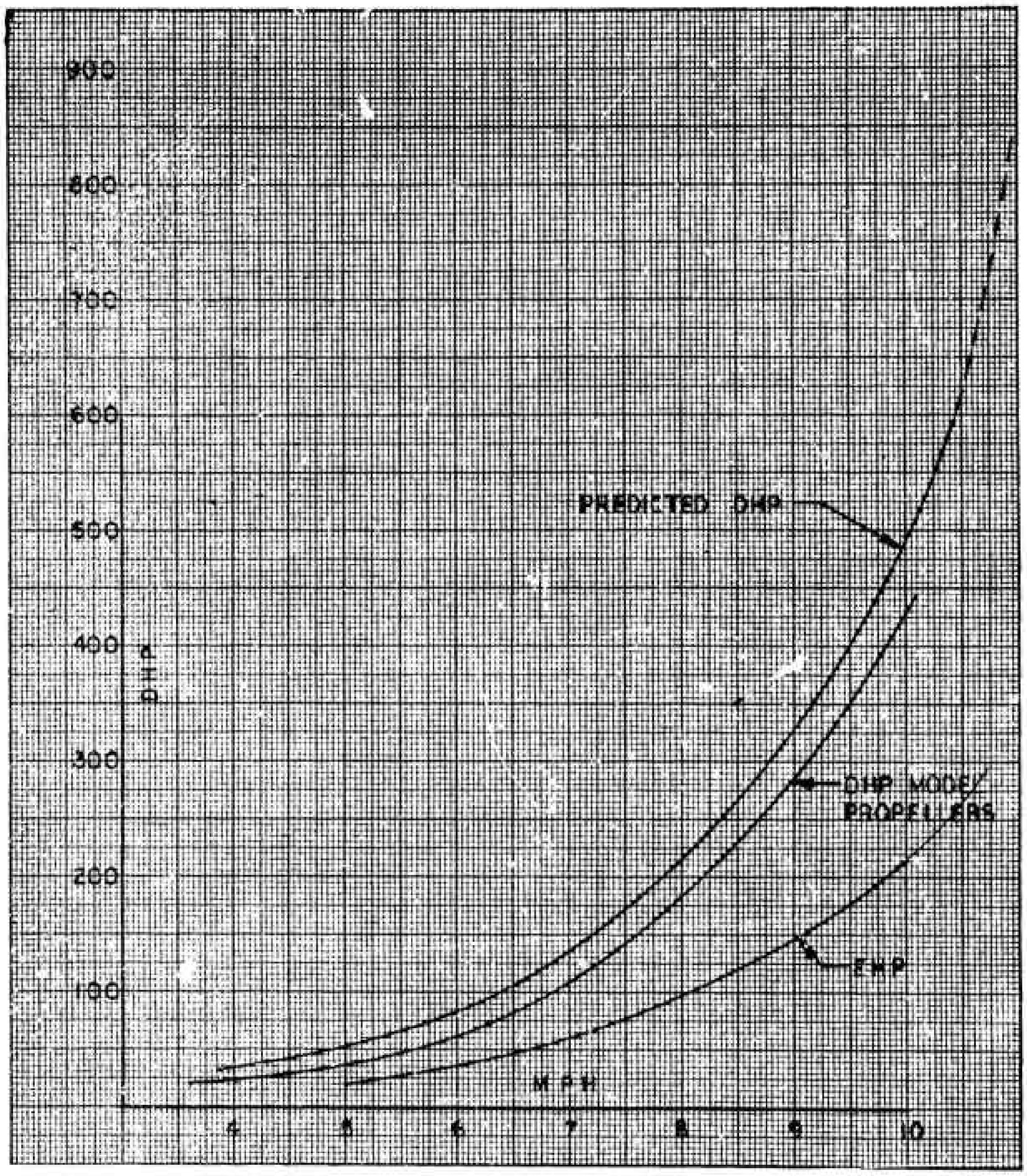


Figure 4-68 Performance of Model Propellers, and Predicted Performance of Prototype Propellers

diameter from 29 to 24 inches, more suitable pitch and blade area ratio would pack back some of this lost efficiency. The resulting efficiency of 24 inch propellers would be in excess of 44 percent.

The curves of model performance in Figure 4-68 are for the vehicle with bow fenders installed. To illustrate the effect of keeping the bow fenders on the LVTPX12, reference is again made to Figure 4-27. In these curves the frictional torque lost in the drive system of the model has not been subtracted from the data, and the curves are for comparative purposes only. They show that retaining the bow fenders requires more horsepower. The curve for predicted DHP in Figure 4-68 is for no bow fenders.

In tests of the model with the propellers retracted into the walls, rotation was such that the pitch times RPM on the model would correspond to the same product on the prototype. The model propeller was immersed to 1/2, 1/4, and 3/4 of its diameter, at displacements of 43,600 and 53,600 pounds. At 53,600 pounds, vehicle speed was 1.1 MPH with 1/4 immersion, and 3° bow trim; 2.1 MPH with 1/2 immersion and .75° stern trim; 3 MPH with 3/4 immersion and 2° stern trim. At 43,600 pounds, speed was 1.2 MPH with 1/4 immersion and zero trim; 2 MPH with /2 immersion and 2° trim by stern. (See Section 24. 0, Appendix A.)

4.4 Height of Bow Wave. At any given speed, the height of the bow wave of the LVTPX12, or the height for any other craft, is inherent in the displacement, the beam, and the bow form. Therefore the speed at which the craft can travel without submerging is fixed by the amount of freeboard at the bow. This varies, of course, with trim. Figures 4-60 to 4-65 show the model being self-propelled by twin propellers. Displacements and trims were

close to those at which the prototype will be under various conditions. The height of the deck at bow for this test was 102 inches from the ground line. The lines on the bow in the photographs are spaced 4.5 inches apart (full scale) and show the height of the wave from the deck in various conditions. From these tests may be obtained a prediction of the maximum speed possible for any hypothetical height of deck. The water in the tank was calm, of course, so that the added effect of a sea will have to be superimposed on these pictures. From these observations the graphs in Figure 4-69 have been drawn. These graphs show that for the vehicle at zero trim and a displacement of 50,000 pounds, the freeboard at the bow cannot be lowered even in calm water for 10 MPH.

At greater vehicle weight, the effect on maximum possible speed is illustrated by Figure 4-70 and 4-71. These photographs were of the model fitted with bow #2 (the round bow) at 59,000 pounds, while the previous pictures are for the boat bow (bow #3) at 50,000, 43,000, and 53,000 pounds. The heavier displacement and the blunter bow combine to limit the speed to 8 MPH at zero trim and 8.55 MPH when the LVTPX12 is trimmed 5 inches by the stern.

The maximum possible speed, however, is not only a function of displacement, but also of trim. Figure 4-69 shows that with stern trim, the LVTPX12 can go well over 10 MPH even at 53,000 pounds. As shown in Section 23, Appendix A, the LVTPX12 when equipped with twin screws is in the favorable region of Figure 4-69 at all conditions of loading.

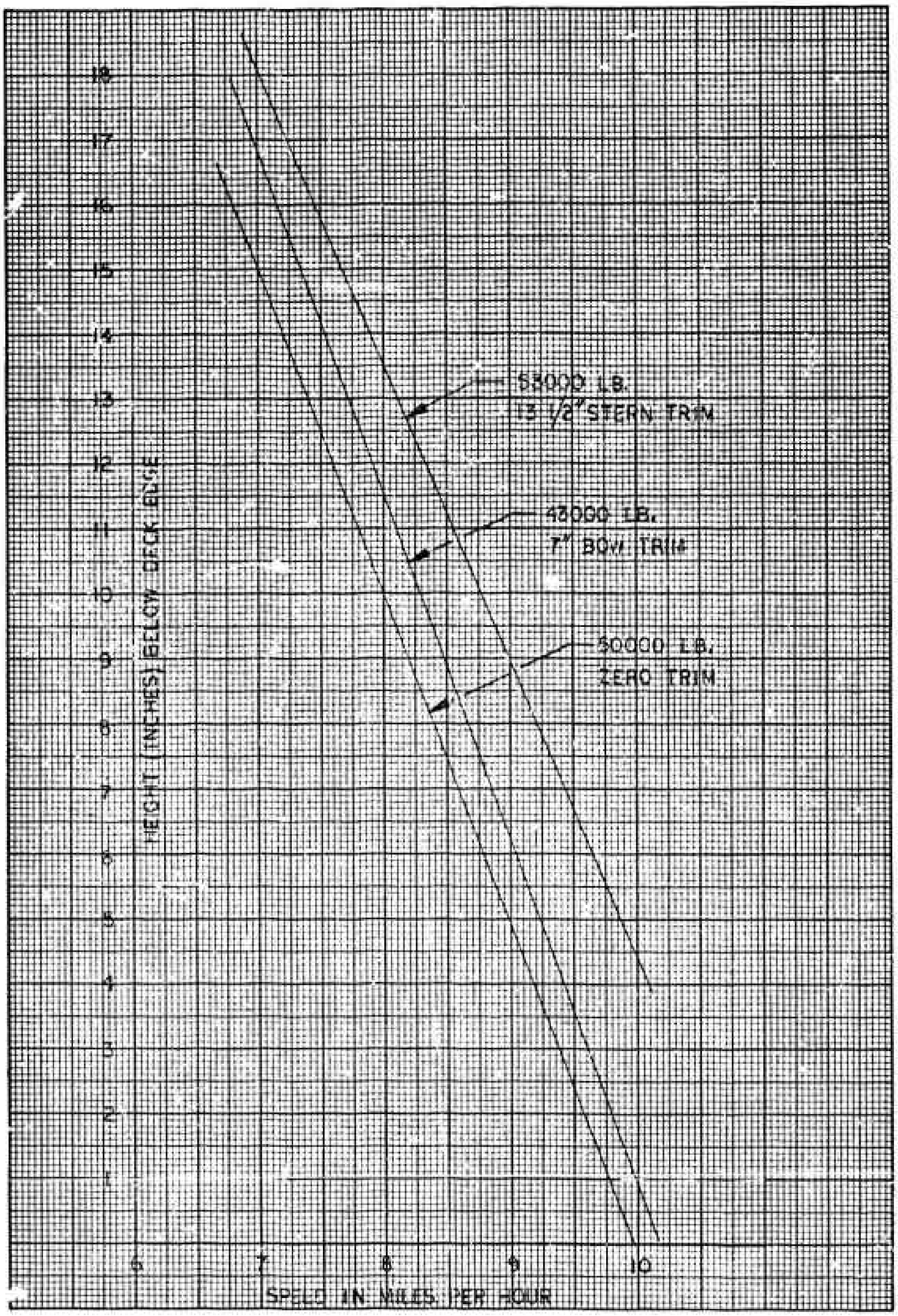


Figure 4-69 Height of Bow Wave Measured From Deck

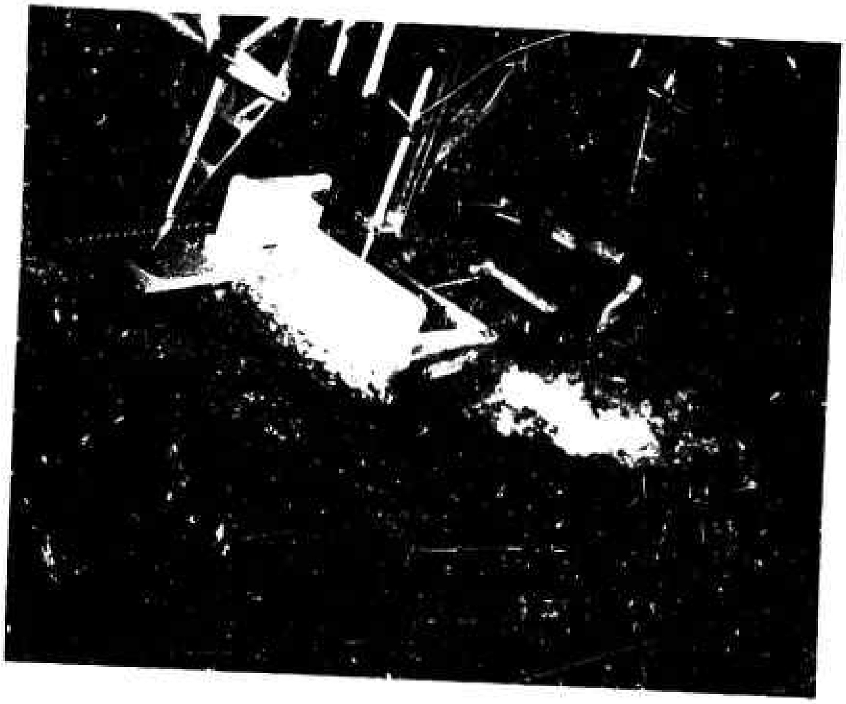


Figure 4-70 8 MPH, Zero Trim, 59,000 Pounds

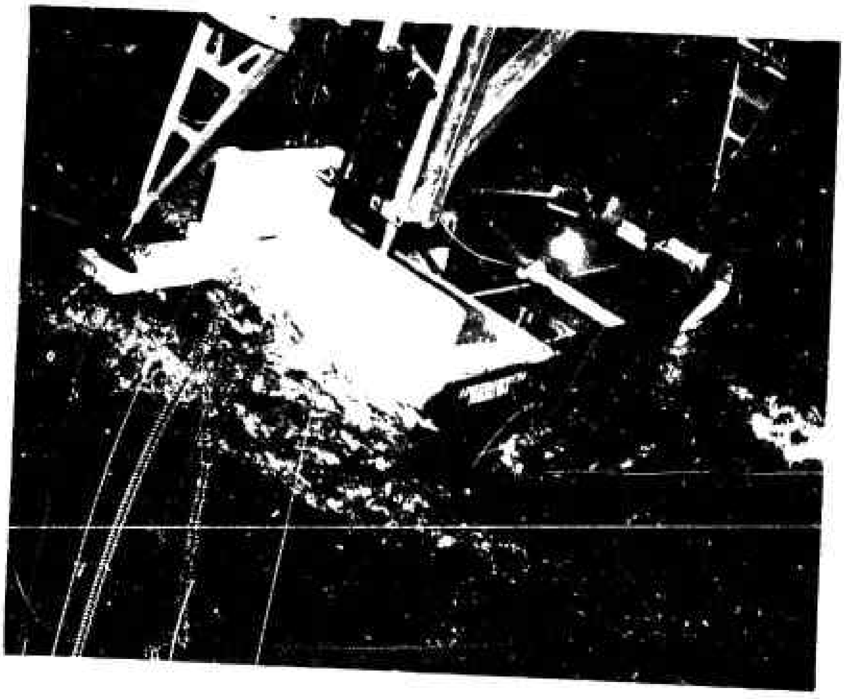


Figure 4-71 8.55 MPH, 6 Inches Stern Trim, 59,000 Pounds

4.5 Conclusions from Model Tests - Screw Propulsion. The best hull for screw propulsion is substantially in this case the same as the best hull for any other kind of propulsion. This might not be true for a vehicle not expected to run at very high Taylor quotients when pushed by either screws or tracks. While the Taylor quotient of 1.37 for the LVTPX12 at 8 MPH is high enough to demand that all the fineness possible be put in the bow, the Taylor quotient of 1.74 for 10 MPH makes that fineness all the more imperative. Bow No. 3, however, is the finest bow that can be designed within the limitations of weight and geometry required for land operation. The boat bow makes anything but a stern ramp difficult if not impossible to design.

It is possible that the craft could be pushed at 10 MPH with a blunter bow, but this would increase the height of the bow wave. The effect of such an increase was shown in Paragraph 4.4. Although the increase in height of the bow wave cannot be calculated for a given bow, it is logical that the adoption of a blunter bow will require either an increase in height or a decrease in speed.

Hence the design criteria for the screw-propelled LVTPX12 are as follows:

- The finest possible bow must be adopted. A stern ramp is therefore practically speaking unavoidable.
- Twin retractible screws mounted at the sides of the vehicle aft are superior to a single screw in most respects: they are less vulnerable; they are more efficient; they can provide thrust when housed; they do not increase the length of the vehicle; they allow a fine bow and a stern ramp, while the single screw does not;

both screws and tracks may be operated together during landing, while the single screw must be retracted before the vehicle hits the bar; in whatever position, they do not place a trimming or heeling moment on the vehicle. On the other hand, they do pose a steering problem, but this is answered by controllable pitch propellers. While the single screw can be used for steering, it will place a large heeling moment on the craft in a sudden maneuver, and this heeling moment can be dangerous if the vehicle is caught at the same time broadside in a steep sea.

- There is no more reason to rake the stern of the screw-propelled vehicle than there is for the tracked one. Nothing will be gained by the little rake that could be put into the stern, but rather some fineness in the bow would have to be sacrificed.
- Although, provided the water is calm, the LVTPX12 when trimmed by the stern with a deck height of 102 inches can go more than 10 MPH, even at the extreme displacement of 53,000 pounds, such happy sea conditions will not often prevail. If 10 MPH is the desired speed, the deck should not be lowered.
- It is possible to design the LVTPX12 so that the operator would have the option of either 10 MPH with twin propellers alone or 8 MPH with tracks alone. Such a vehicle must be equipped with large bow fenders, stern baffles and short contravanes. At the same time, the resistance would be somewhat increased. Without these appendages, the speed will be slightly in excess of 6 MPH by track propulsion, if the water grousers are retained (grouser #1). The power for screw propulsion can be reduced still further,

of course, by adopting a grouserless track, but water speed with track propulsion would be very low. (See Section 9.0). Therefore, the water grousers should be retained. But the most economical performance in the long run can be obtained by omitting the appendages and accepting 6 + MPH by tracks or 10 + MPH with propellers, at the option of the operator. Thus the LVTPX12 could maneuver under any conditions of trim, and could make better speed on tracks alone than most of its predecessors.

Hence the optimum shape of the screw propelled LVTPX12 is barely different from the one with track propulsion. A lines drawing of the best track propelled LVTPX12 is shown in Figure 4-72.

4.6 Steering and Maneuvering. Tracked amphibians have been characterized by operators as inherently unstable on course. Besides the slower net speed resulting from a zig-zag course, steering by tracks results in decreased propulsive efficiency when one track is slowed down, even though the opposite track might speed up. Hence, for track propulsion, some more efficient means of steering is worth seeking. Rudders for the track driven vehicle are so obviously impractical that they will not even be considered. The propeller driven vehicle itself presents a challenge in steering, and if the propeller installation is to yield maximum benefit, it must result in extraordinarily good maneuverability.

4.6.1 Steering of Track-Propelled Vehicle. Rudders being out of the question, and track steering inefficient, the best method of steering the LVTPX12 is by some means of reactive steering by power. This can be accomplished either

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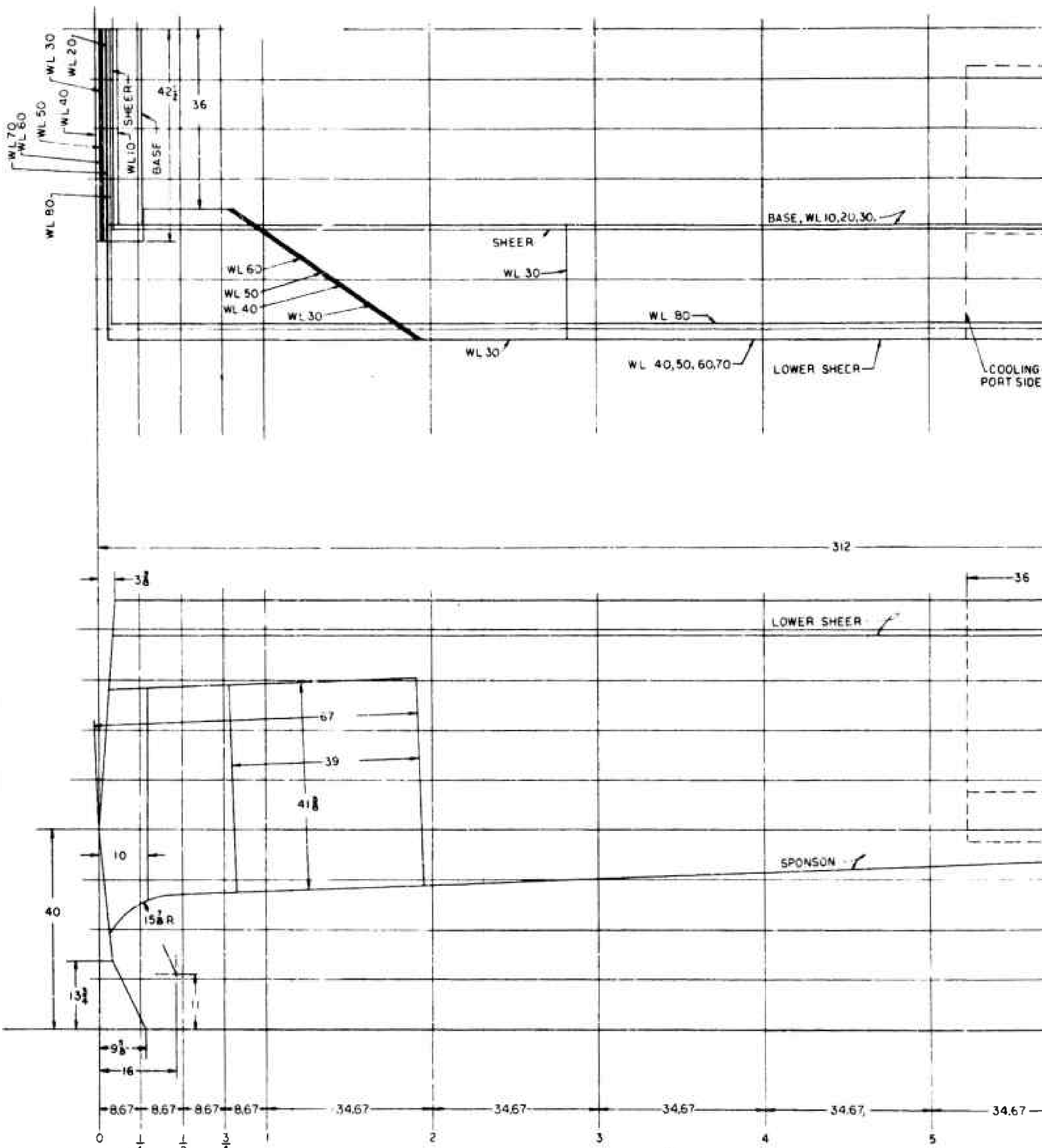

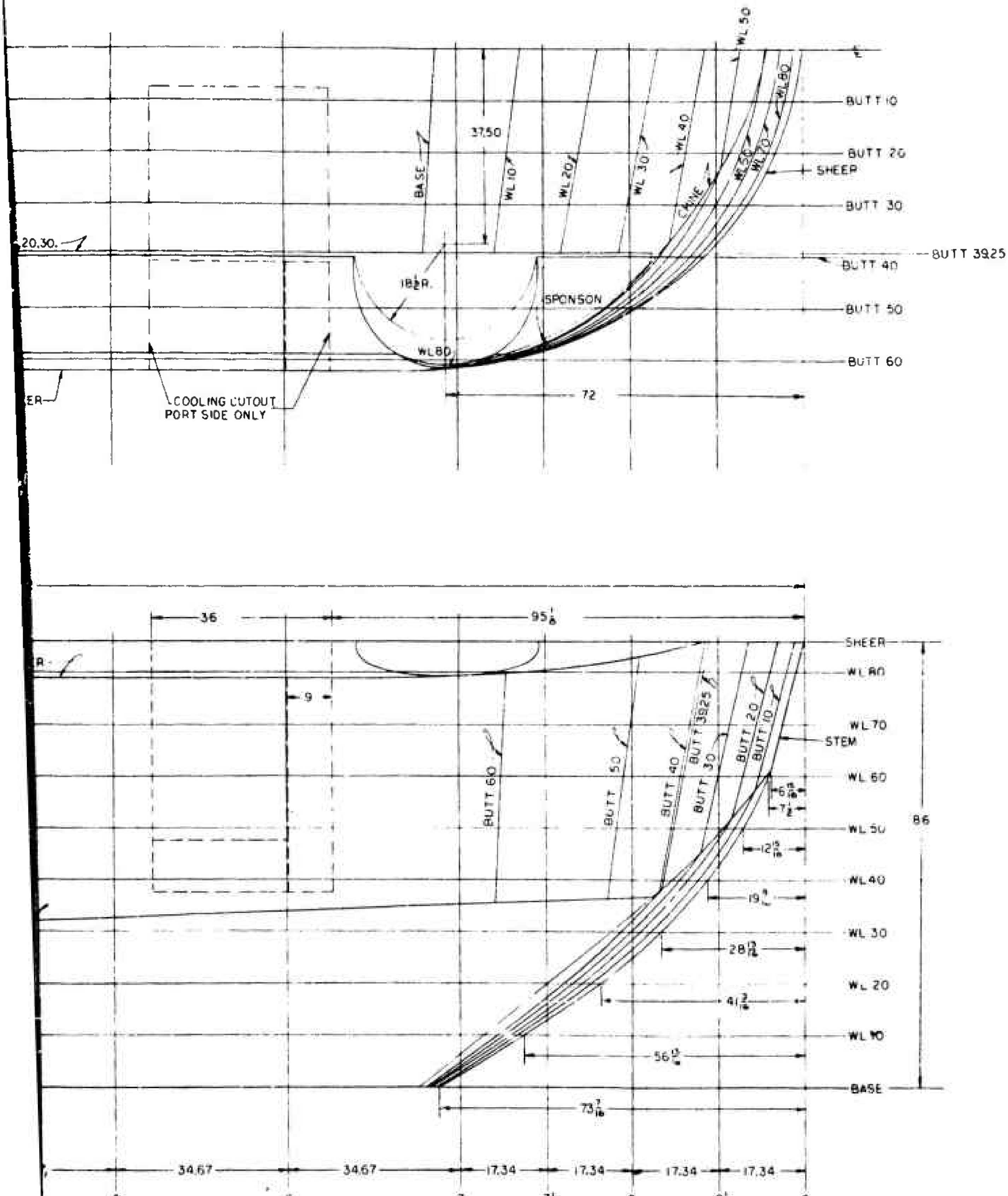

WL 50
BUTT 10
BUTT 20
SHEER
BUTT 30
BUTT 39.25
BUTT 40
BUTT 50
BUTT 60
37.50
BASE
WL 10
WL 20
WL 30
WL 40
CHINE
WL 60
WL 70
WL 80
20.30.
18½R.
SPONSON
72
COOLING CUTOUT
PORT SIDE ONLY
36
95⅛
9
STEM
BUTT 60
BUTT 50
BUTT 40
BUTT 39.25
BUTT 30
BUTT 20
BUTT 10
86
BASE
34.67
17.34
5
6
7
7½
8
8½
9

2

925

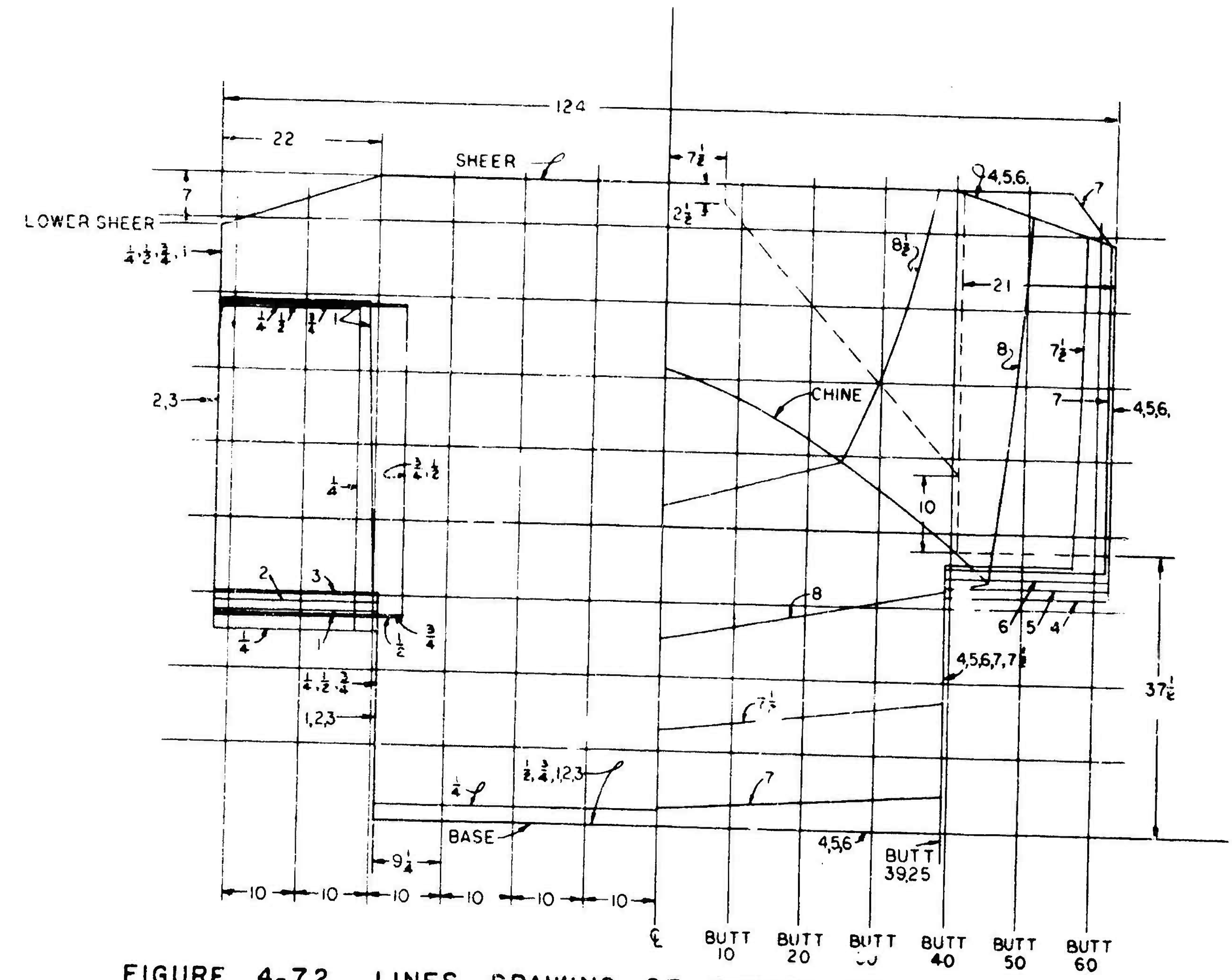

FIGURE 4-72 LINES DRAWING OF PROPELLER DRIVEN VEHICLE

by side thrusters or by small jets on either side at the stern. The design of such steering aids is beset by unknown requirements as well as by unknown effects. The specifications call for "maximum attainable" maneuverability, but the degree of maneuverability that should be sought is not known. The specifications do not mention coursekeeping. No model tests have been conducted to determine the coursekeeping qualities of the LVTPX12 nor to determine either the response of the vehicle to a given turning moment or the amount of turning moment that could be supplied by any particular means of steering. Such tests would be complicated, lengthy, and expensive, and moreover, would be of uncertain value at this time because no such model tests have ever been made for later comparison with prototype behavior.

A side thruster on the track-propelled vehicle would necessarily have to be located in the bottom, beneath the floor, because of the rear ramp. Its ejection would be into the tracks and therefore ineffective. A pair of small jet pumps, however, can be placed on the sponsons above the tracks on either side and will not interfere with passage in the cargo compartment. Such a pair of 7.5 inch jets would occupy only 39 inches along the sponsons at the stern of the vehicle, and would weigh 87 pounds each, not including the driving machinery. The total resulting increase in weight, however, is estimated at 1100 pounds. (See Appendix A, Section 12.0). If electrically driven, the jets could supply a continuous variation of turning moment, both by shifting power from one side to the other and by use of directional nozzles, at the command of the operator. (See Appendix A, Section 12.0). At some point, as illustrated in Figure 4-73, it should be most profitable to invest part of total available engine power in such a steering device and the balance in track propulsion. The ordinate in Figure 4-73 stands for actual speed

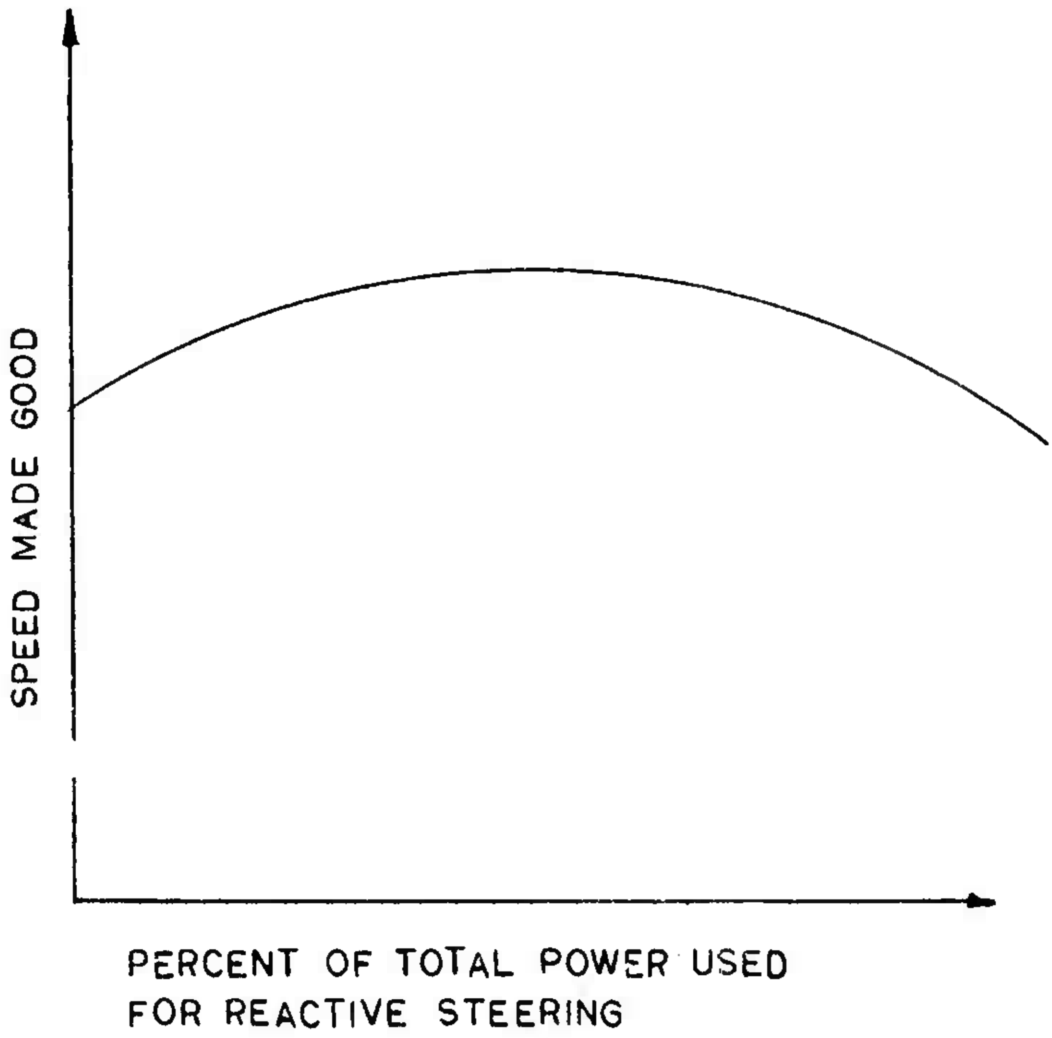

Figure 4-73 Probable Optimum Speed Made Good

made good, not rated speed for the vehicle forced by lateral restraint to stay on course. The value of the abscissa for the maximum point of effective speed is not known. The only way to find this point is by tests of the prototype.

Because of the weight of such a device and the impossibility of subjecting it to model tests, it should not be adopted in this stage of design, but it should be tried on one of the prototypes in Phase II.

The reason for lack of stability on course of any craft is a center of lateral resistance forward of the center of gravity. A source of high forward pressure in conventional amphibians is a large vortex alongside, just abaft a blunt bow. As is observable from the photographs of the model with the boat under way at high speeds, this vortex is largely absent in the case of the LVTPX12. Thus the LVTPX12 can be expected to hold its course more readily than blunt-nosed amphibians.

4.6.2 Steering the Propeller-Driven Vehicle. The LVTPX12 can hardly be expected to go straighter under propeller drive than under track propulsion. Small rudders are possible with the twin screw concept, but they represent only one of a long list of possibilities:

- Rudders, either split shrouds, ring shrouds, or streamlined trailing vanes.
- Clutches for throwing out one propeller at a time, but with no reverse gears.
- A controlled differential mechanically providing continuous variation in speed between propellers but not full reverse on one screw with the other rotating ahead.

- Separate and independent control of propeller speed by hydrostatic transmission.
- Steerable propellers with a limited angle of swing, but not independently reversible nor independently variable in speed.
- Steering jets or side thrusters, providing little reverse thrust or none at all. (Appendix A, Section 12.0).
- Independent drive from twin engines through reverse and reduction gears, speed differential controlled by varying throttle on engines.
- Independent reverse gears, but with no independent control of speed.
- Electric drive with independent control from full ahead to full astern.
- Controllable pitch propellers, independently actuated from full ahead to full astern.

The first difficulty with rudders is that they increase the length of space devoted to the outboard drive. When housed they must not protrude past the stern of the vehicle. Supposing two extra feet of length of the propeller well were allowable, the rudders would still provide only limited maneuverability at slow speeds.

The manifestly objectionable characteristics of many of the other means in the above list make them hardly worth discussion. The steerable screw idea, however, would provide high maneuverability and good steering.

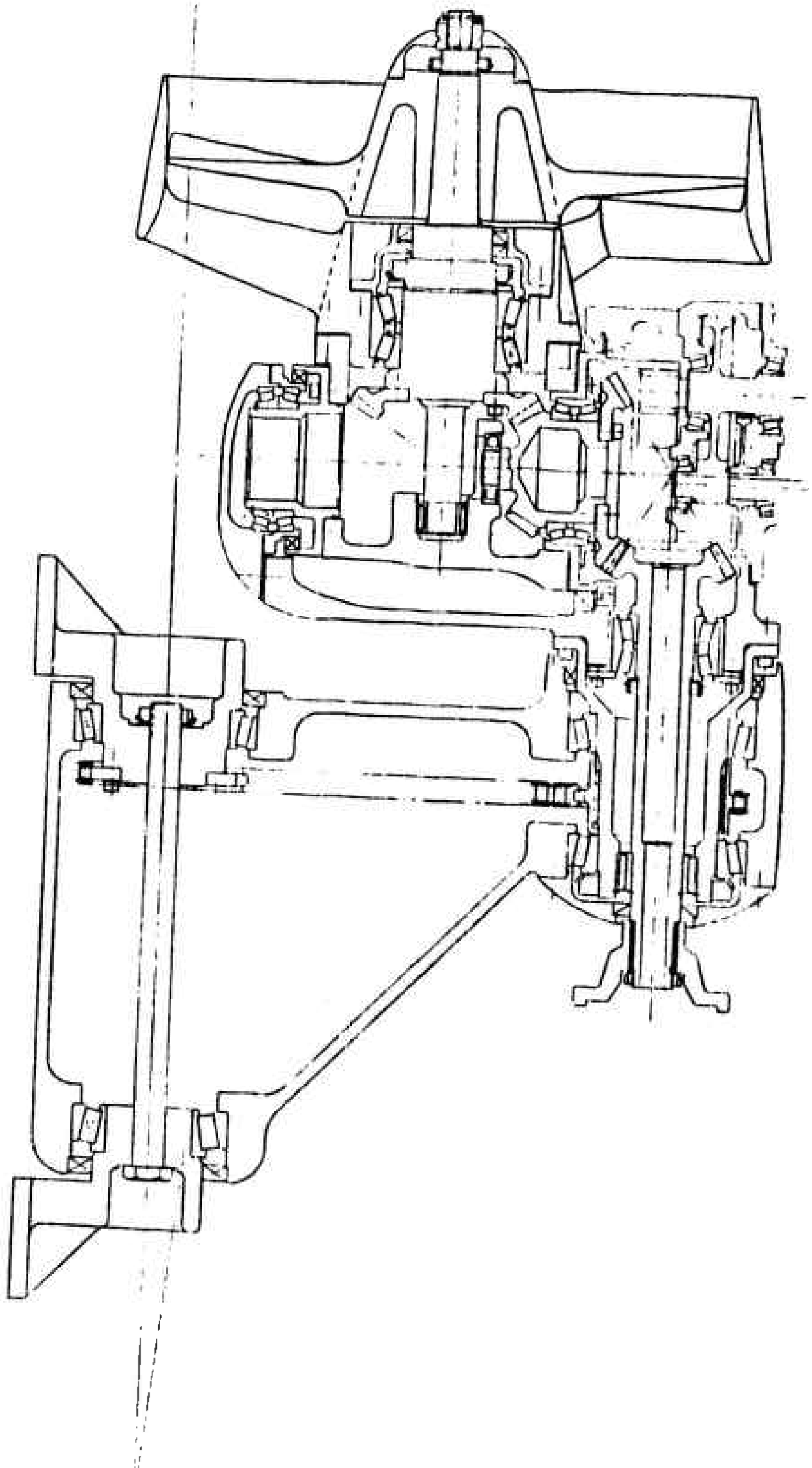

Figure 4-74 A Steerable Outboard Side Propeller

As in the case of the rudders, the steerable propeller would have to be reversed or slowed by a complicated differential drive system.

An illustration of a steering outboard drive is shown in Figure 4-74. The design incorporates a simple means of maintaining a vertical rotational axis throughout the retraction arc to provide maximum horizontal thrust at all positions. Without going into the question of reversing or independently controlling speed, it is evident at once that the housing of the gearing and controls would be almost as broad as the screw itself. The resulting inefficiency eliminates this system from further consideration.

With the exception of independent electric drive and controllable pitch for steering and maneuvering the twin screw vehicle, the effectiveness of the others would be poor if they were effective at all. Their cost, weight, and complexity would make them eligible only as a last resort. Steering by either the electric drive or by independent controllable pitch, however, would be positive and immediate. The wide range of differential thrust would make the vehicle steer like a boat, and would perform any desired maneuver, especially in close quarters. Moreover, on mounting a beach or crossing reefs, the tracks could turn at any desired speed with the propellers engaged. Throwing power to the tracks would not load the engine down and reduce engine power, because the operator could decrease the load on the propellers at will by a simple change in pitch.

Electric drive is out of the question unless the entire propulsive system is to be electric. Controllable pitch propellers provide even more positive and immediate response than electric drive. No brakes, nor waiting for energy

to be expended, are necessary for reversing. The propellers may turn at constant speed, and pitch can be changed from full ahead to full astern in five seconds. At any engine speed, the pitch can be set for optimum efficiency. As the resistance of the craft changes due to changes in load, the pitch of the propellers can be matched to the changed condition.

Although the efficiency of a controllable pitch propeller is inherently a little less than that for fixed pitch, where the designs are both for the same operating point, the adaptability of the controllable pitch to changed conditions or to conditions not anticipated in the design makes its efficiency in the long run a little better. Controllable pitch screws have been popular in Europe for many years, and have been recognized in this country as excellent for installations where loads are heavy and operating conditions are variable. They are supplied by a number of manufacturers. Their reliability is soundly proved.

4.6.3 Operation of Controllable Pitch Propellers. Of the many perfected controllable pitch propellers being manufactured, all operate in the manner schematically illustrated in Figure 4-75. The pitch change is effected in all of them by a simple device inside the propeller hub. A number of patents are held by domestic and foreign owners on cam designs for the changing of pitch while the propeller is turning at full speed, but all of them are actuated by a control rod moving short distances back and forth. The control rod is moved by a linear actuator either in front of the propeller or behind it. The linear actuator is operated either electrically or hydraulically by remote control.

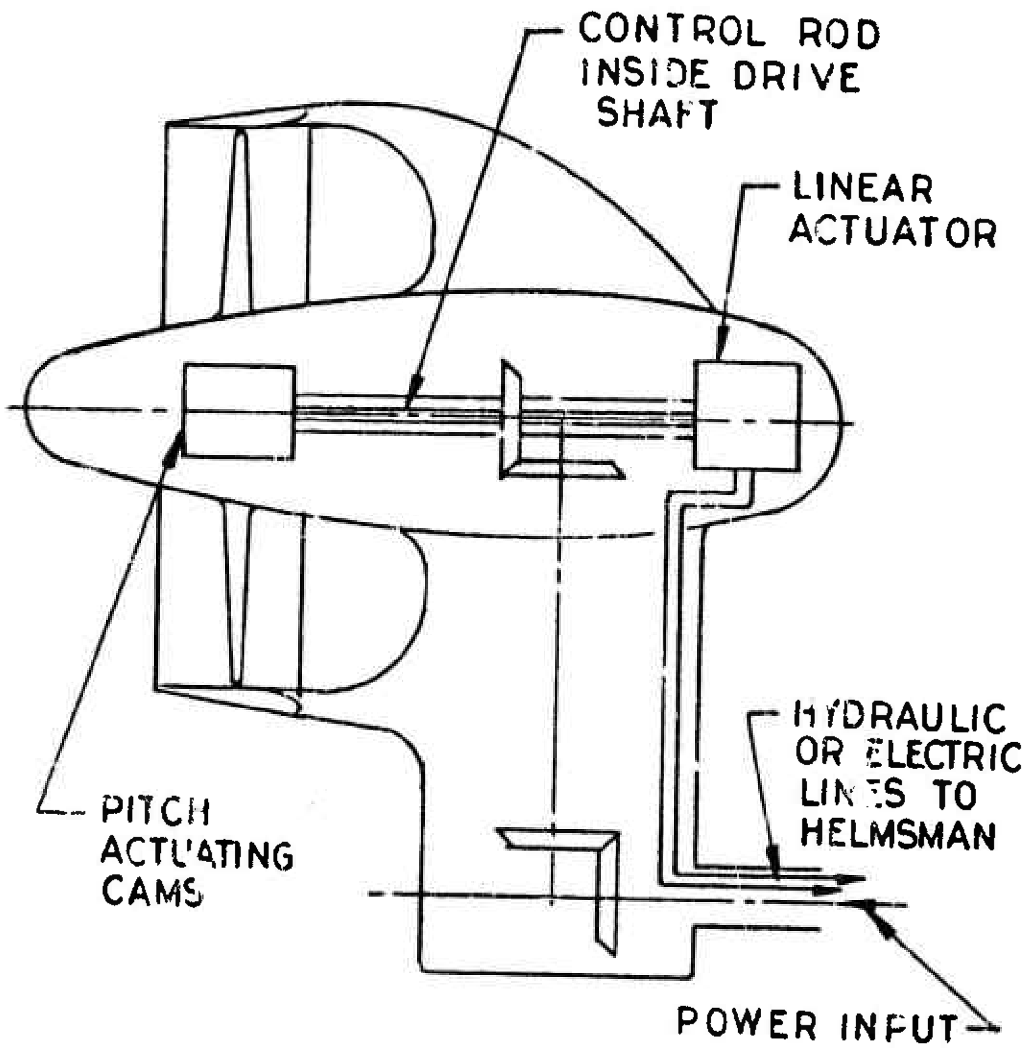

Figure 4-75 Essentials of a Controllable Pitch Propeller

One of the features emphasized by most manufactures is the quick removability of blades and the ease of maintaining and repairing units. The reason for development of these features is that especially in the United States, where the advantages of controllable pitch have become widely recognized only since 1950, prospective buyers have often expressed fear that there was something peculiarly vulnerable about a submerged mechanical device. Even though these fears have been ungrounded, the special effort of manufacturers to answer them must be acknowledged a benefit. (See Section 15, Appendix A.)

Essential to the control of propeller pitch is a means whereby the captain of the ship can order an exact pitch and know that the order is executed. Such a requirement is no more difficult to meet for a propeller than for a clutch, a throttle, rudder, or any other service where exact movement must be produced by signal from a distance. Whether the power for actuation is electric or hydraulic, the operator moves a simple lever to obtain the desired response, just as in the operation of a clutch control or throttle. For independent control of two propellers, two such levers are used.

The response to movement of these levers begins almost instantaneously when the signal is initiated, but naturally a time lag occurs in completing the response, depending on the amount of pitch change ordered in the signal. If pitch is set on full ahead and the signal is to change pitch to full astern, for example, completion of the response takes twice as long as it would if the signal were to change pitch only from full ahead to neutral. The linear actuator can be built to provide any reasonable speed of response. On the LVTPX12 a response time of five seconds from full ahead to full astern should be adequately short. Small changes of pitch would require corres-

pondingly short periods.

For the simple exercise of steering a straight course, the operator would soon learn that he should apply turning moment for course correction by gradual change of pitch rather than by sudden large movement, just as he would in steering any other vehicle on land or sea. The response of the LVTPX12 to the helm will depend on the amount of turning moment applied. Small changes of pitch will produce enough moment to correct deviations from course, as illustrated in Figure 4-76. Very large turning moments will cause the craft to overshoot the course. In this respect steering the LVT will be like steering any other seagoing craft.

In most maneuvering and coursekeeping, the engine will be kept running at constant speed. The propellers may be operated, of course, at any chosen speed of rotation. If tracks are engaged while the propellers are also engaged, the engine will not be able to supply full power to both. Since the speed range of tracks is not continuous, but depends on the gear ratio, the operator will have to reduce propeller pitch when both tracks and propellers are turning.

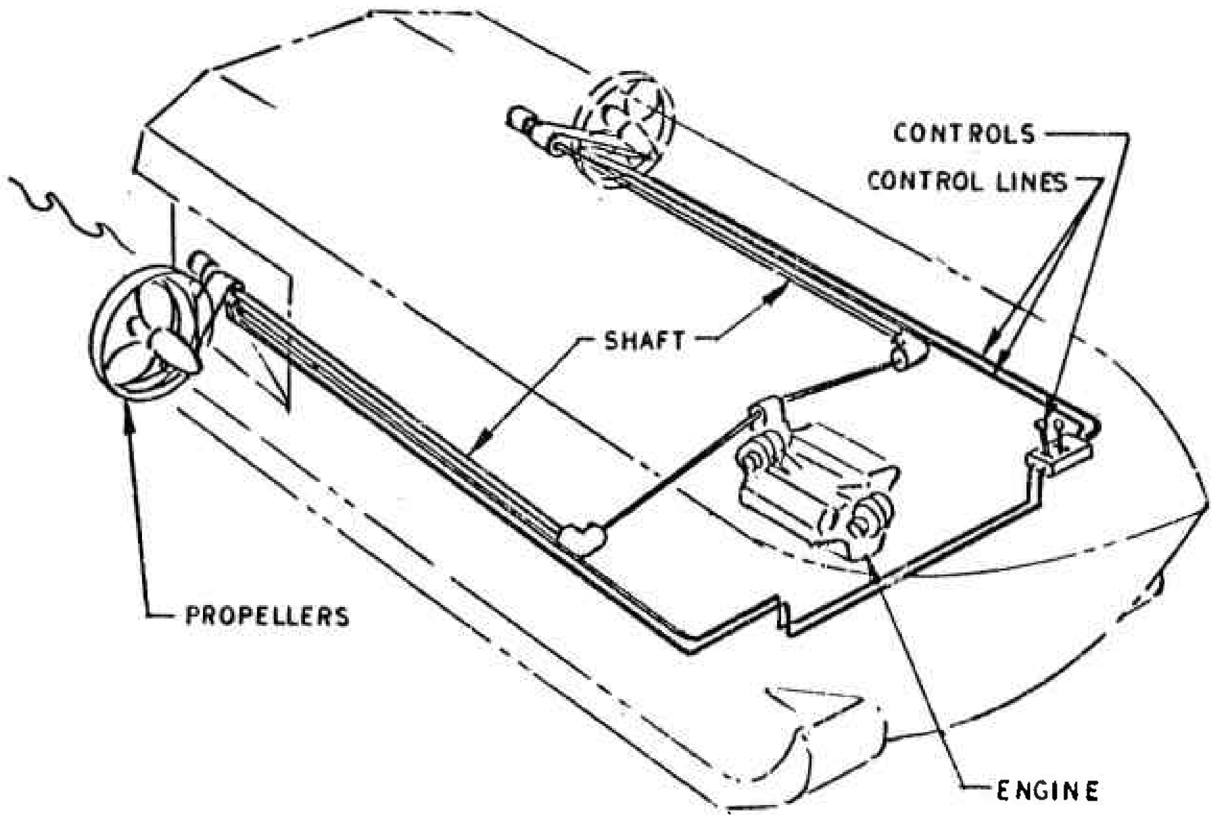

Figure 4-76 Steering by Pitch Control

4.7 Problems Meriting Further Research To repeat the observation made at the beginning of Paragraph 4.2, not every lead that appears worth following can be followed in a single program having a time limit. The model testing program of the LVTPX12 has not only yielded a low demand for delivered horsepower, but it also has raised some questions that need investigation for the promise of still better performance.

- The paradox of lower efficiency with wider grousers (See Paragraph 4.2.4), is not adequately explained. Rationally, increasing the width of the grousers should increase efficiency. What needs to be done to allow the rational prediction to come true?
- Is there any practical way to supply more water to these turbine blades of the tracks, thus increasing the output? If there is, the answer may be found to the question about the wider blades.
- How can the directional stability of tracked amphibians be improved?
- Tracks being inherently such inefficient propulsive devices, is there some mixture of track and jet propulsion resulting in greater speed at fixed invested power, without prohibitive penalties in weight, space, or complexity?
- In what proportions is the total resistance of a box-like body composed of form drag, wave drag, and frictional drag? It will take some time to accumulate the knowledge for the answer to this question. Experiments of a fundamental nature will have to be performed. The test data from all models and full scale vehicles will have to be comprehensively documented.

4.8 Static Particulars of the Water-Borne Vehicle. Displacement and other curves for each of the two concepts are included on single sheets in Figures 4-77 and 4-78. Curves of stability are shown in Figures 4-79, 4-80, 4-81, 4-82, and 4-83. An additional set of curves for a narrow vehicle of 104 inch beam may be found in Appendix A, Section 18.0 sheet 33.

4.8.1 Displacement and Other Curves. For any chosen draft, the particulars of the screw propelled vehicle may be read off from Figure 4-77, and for the track propelled vehicle from Figure 4-78. These curves provide any desired information on the static characteristics of the craft. The position of each weight used in these calculations may be found in Section 23.0, Appendix A. Movement of weights from these positions will result in a change of trim, and particular attention is called to the hypothetical positions of cargo. The positions used for cargo may be found in Appendix A, Section 23.0, but in case cargo is concentrated at some other point the resulting change of trim may readily be calculated by consulting the curves in the manner to be described and finding the moment to trim one inch.

As an example of the use of these curves, let the track propelled vehicle be loaded with 6,810 pounds of cargo, bringing the gross vehicle weight to 48,800 pounds. Instead of loading this cargo so that its center of gravity is 92 inches from Station 0, let the center of gravity be a distance d away from that point. Divide 48,800 by 8,000 and obtain 6.1. This is the point at which the scale at the bottom of Figure 4-78 will be entered. Thence at the intersection of the 6.1 inch distance with the displacement line, read horizontally:

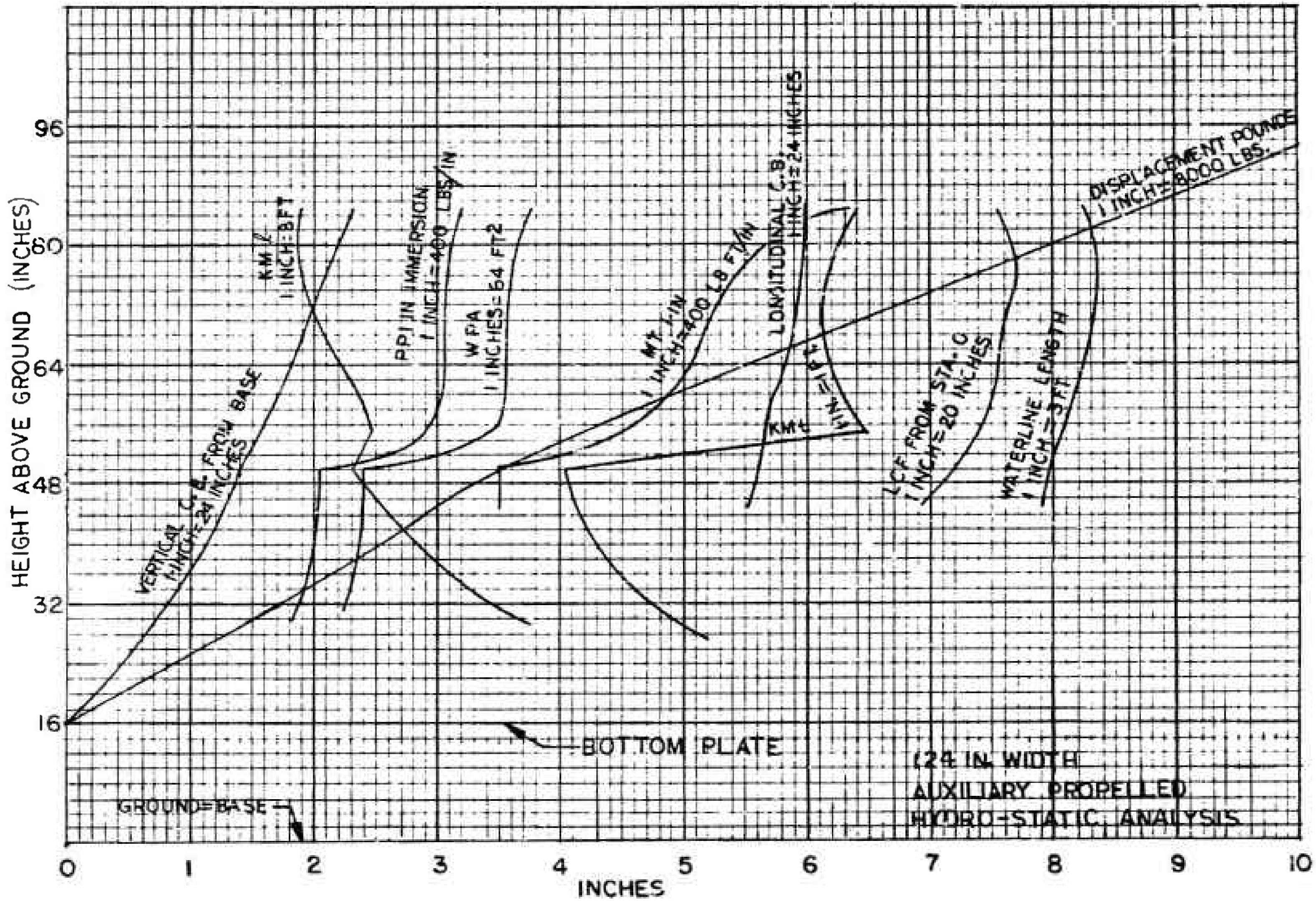

Figure 4-77 Displacement and Other Curves For Propeller Driven LVTPX12

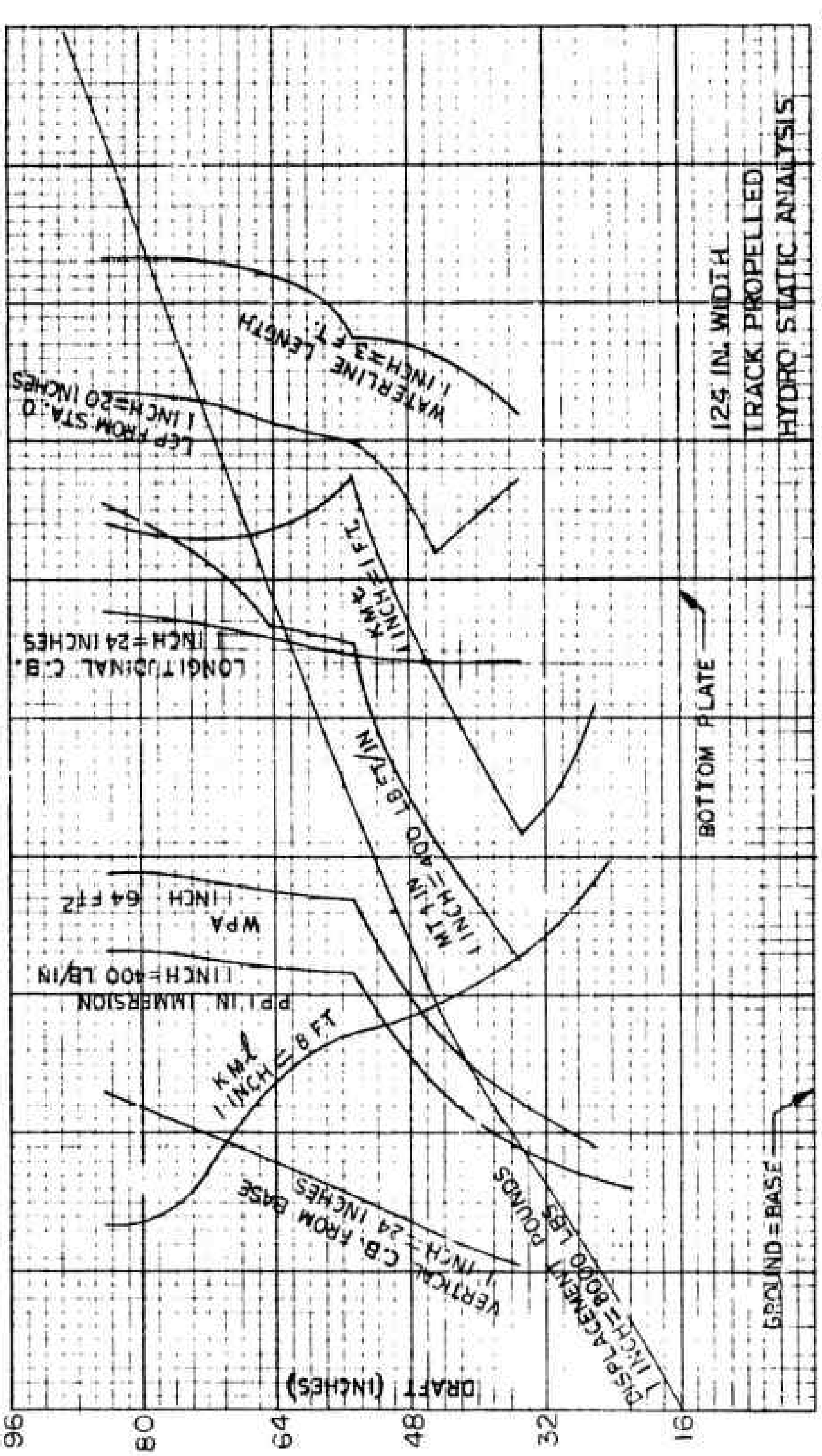

Figure 4-78 Displacement and Other Curves For Track Propelled LVTPX12

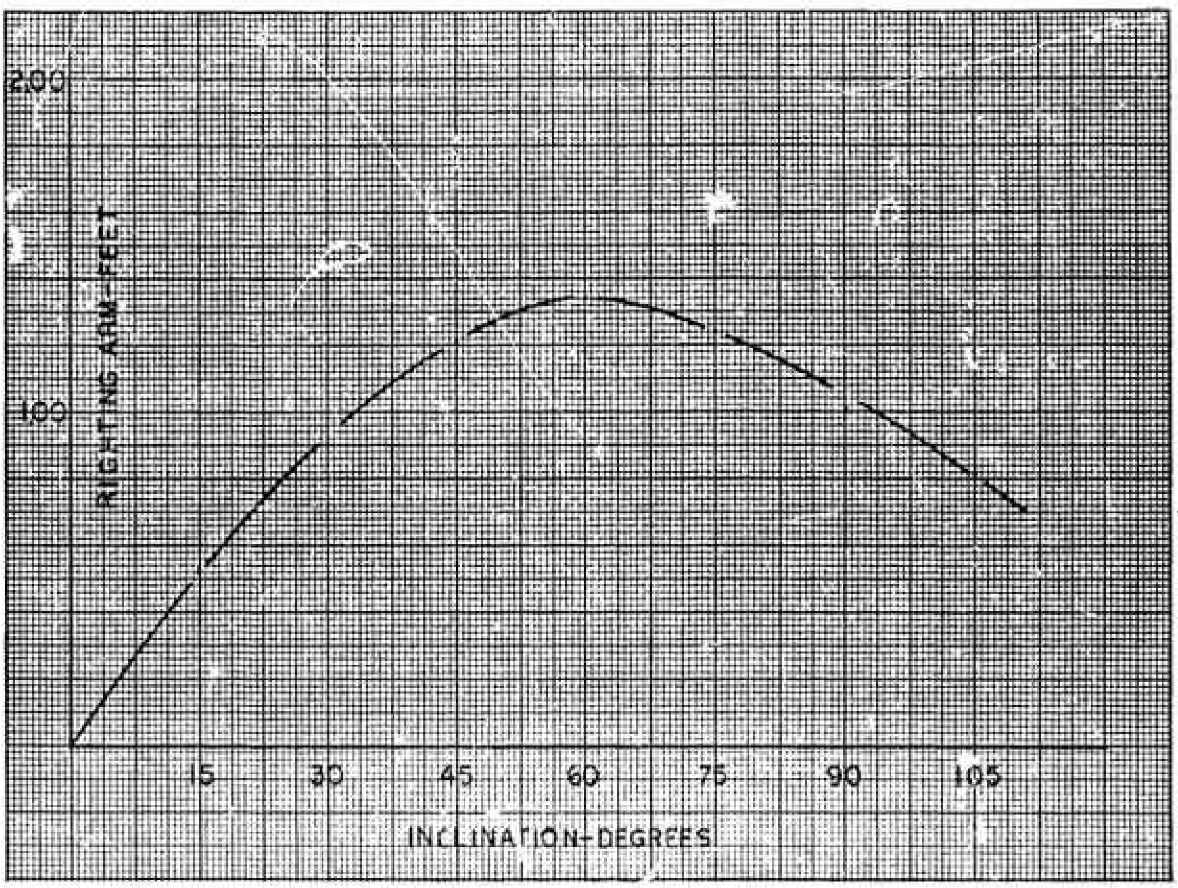

Figure 4-79 Righting Arms For Track Propelled LVTPX12 At 41,380 Pounds

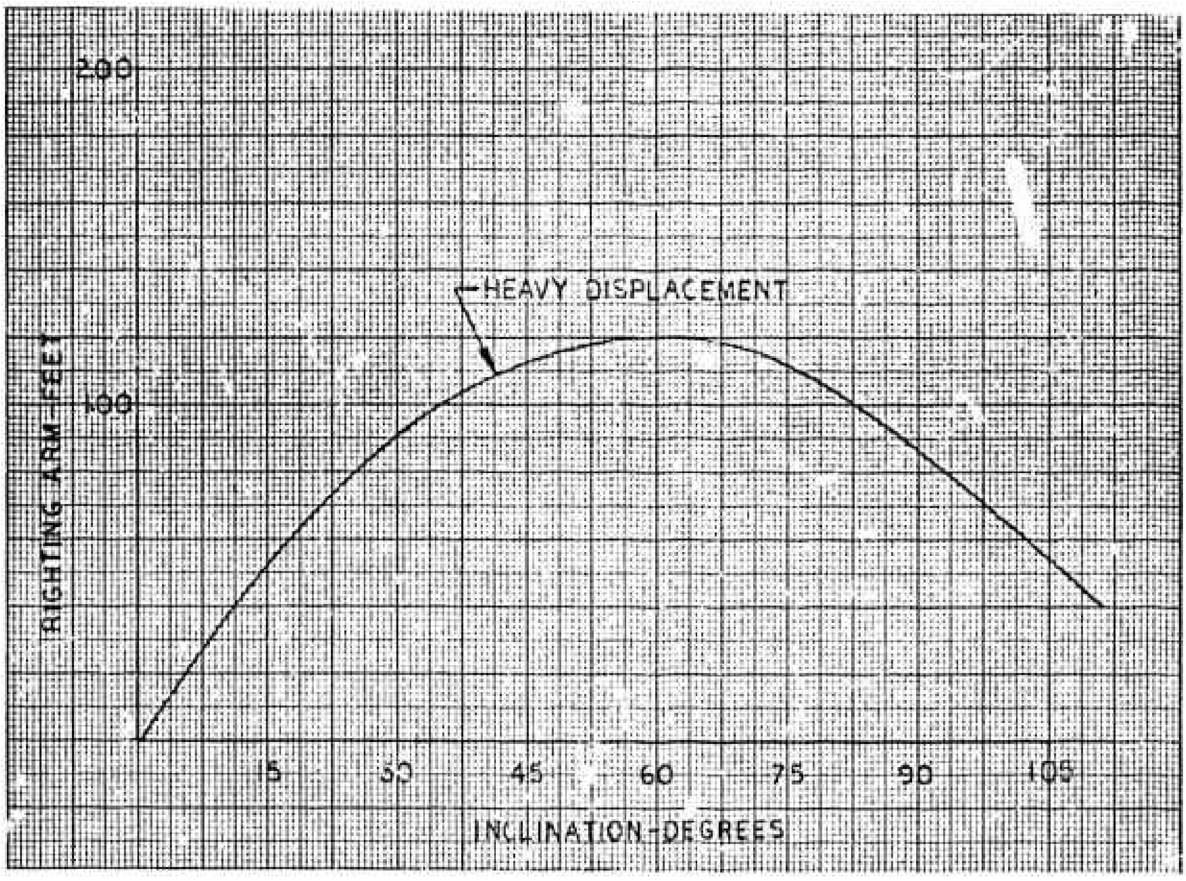

Figure 4-80 Righting Arms for Track Propelled LVTPX12
Fully Loaded, 51,990 Pounds

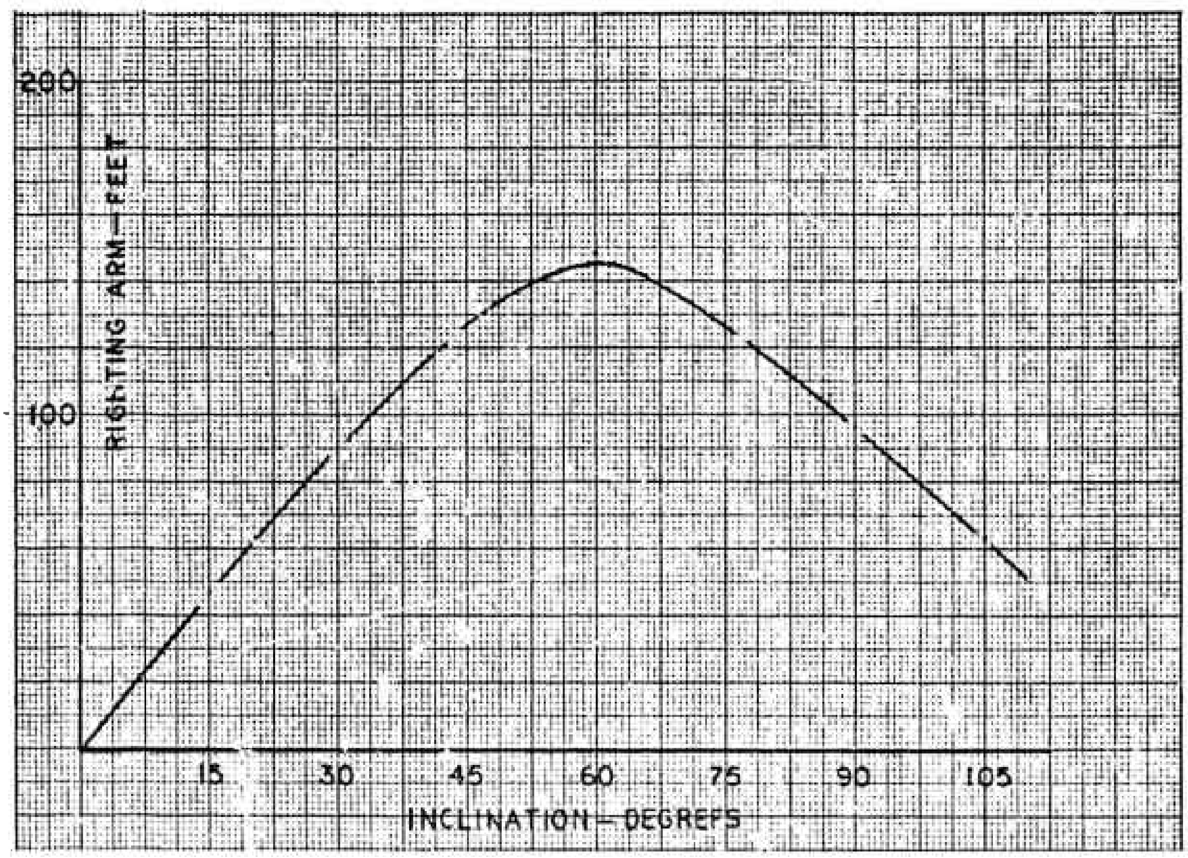

Figure 4-81 Righting Arms, Screw Propelled LVTPX12, Light Condition, Full Fuel, 41,655 Pounds

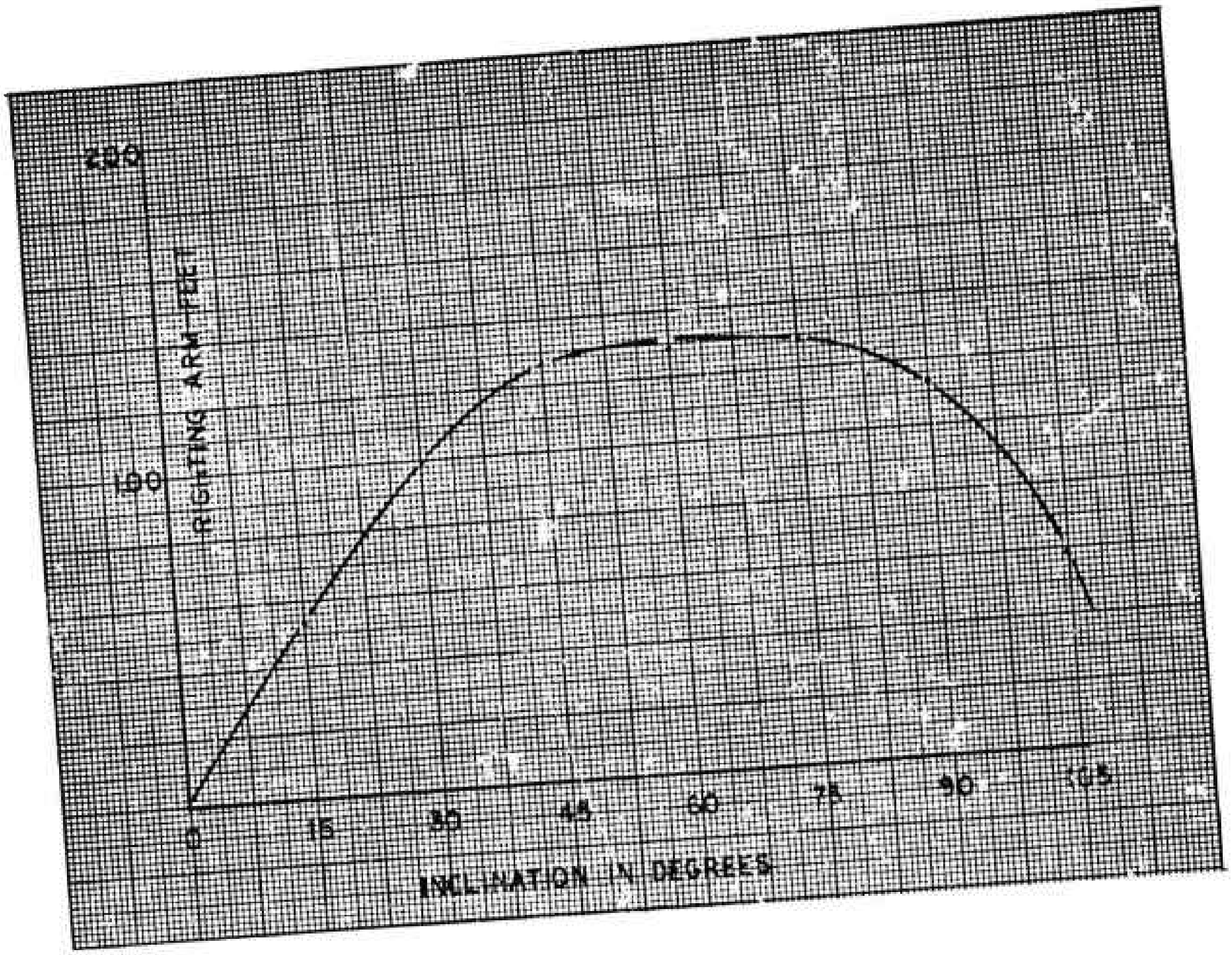

Figure 4-82 Righting Arms for Screw Propelled LVTPX-2, at Displacement of 51,655 Pounds

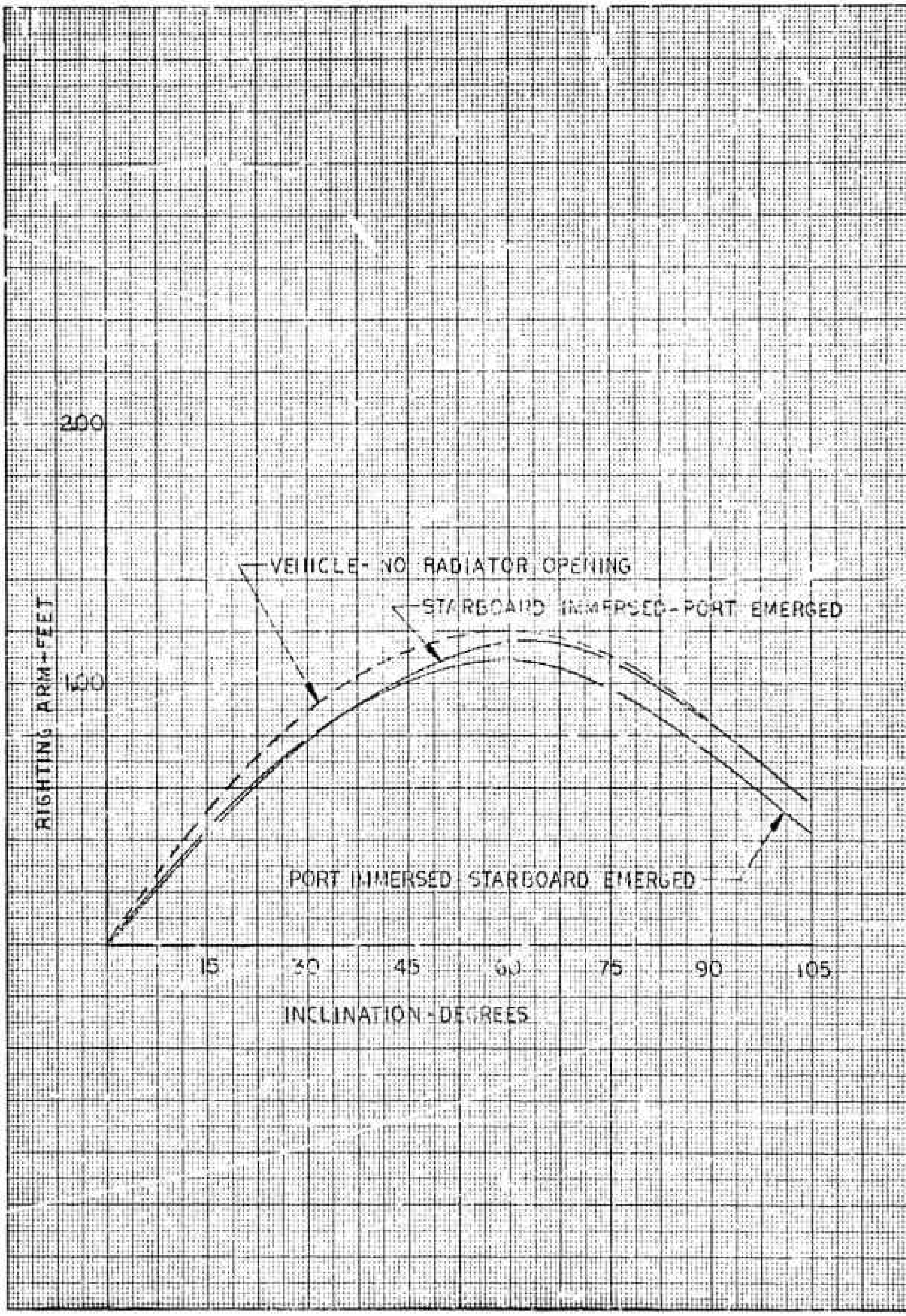

Figure 4.83. Righting Moments Showing Effects of Radiator Well

Draft	66 inches
Vertical CB from ground	43.6 inches
Longitudinal KM	17.6 feet
Pounds/inch immersion	1270 pounds per inch, salt water
Waterplane area	241 square feet
Longitudinal CB	134 inches from Station 0
Moment to trim 1 inch	2300 pounds-feet/inch
Transverse KM	6.3 feet
Longitudinal CF	143 inches from Station 0
Waterline length	296 inches = 24.7 feet

Abbreviations above: CB - center of buoyancy; KM - metacentric height measured from base (ground) line; CF - center of flotation of waterplane. The ground line at bottom of track has been used as the base line in all calculations.

Now a weight of cargo, w, has been moved a distance, d, from the reference position. The resulting change in trim is

$$\text{trim} = \frac{wd}{MT1''}$$

The moment to trim one inch, as well as all other particulars, must always be obtained at the proper draft for the given displacement.

4.8.2 Righting Moments. The stability of the LVTPX12 when heeled is given for the following conditions:

- Track propelling, light with full fuel, Figure 4-79.

- Track propelled, fully loaded, Figure 4-80.
- Screw Propelled, light with full fuel, Figure 4-81.
- Screw propelled, fully loaded, Figure 4-82.

In calculations for these curves, the volume of tracks and suspension system has not been included in immersed and emerged wedges. The effect of this volume would be hardly measurable. The effect of free surface in slack tanks has not been included, but the moment of inertia of surface in tanks is extremely small in proportion to the total waterplane. The righting arm at extreme heel is so large for all displacements that an ample margin would remain after any such small corrections. These curves are not carried to the extreme range of stability, and they should not be extrapolated to zero righting arm, unless only a rough estimate is desired.

The radiators being in one side of the vehicle, this void produces a difference in the righting moment from one side to the other. Figure 4-83 shows the curve of righting arms for the track propelled vehicle, fully loaded, when heeled to the radiator side. Because of a slight difference in displacement, the righting arm is a little greater on the radiator side than on the other side at low angles of heel. The calculations have been made only for the heavy condition of the tracked vehicle, and not at all for the screw propelled version, because the effect of the recess is shown to be negligible.

All calculations for righting moments are included in Appendix A, Sections 18.0 and 19.0. The areas and moments of immersed and emerged wedges were measured by integrator.

4.8.3 Static Trim. Calculations for centers of gravity and trim for the vehicle when powered by the GM 12V71T engine, both for the track propelled and the screw propelled versions, are found in Appendix A, Section 23.0. Additional studies are included for the screw propelled LVTPX12 powered by the GM 8V53T and the GM 8V71T engines. These lighter engines do not produce a large difference in the static trim. No smaller engines are included in the calculations for the track propelled version, because they would not drive the vehicle at the specified speed of 8 MPH. Although smaller engines would drive the screw propelled version at more than 8 MPH, they would not reach 10 MPH.

In the following summary, "light" means without cargo or passengers but fully fueled and equipped, with full crew of three. "Troops" means that 25 passengers with combat equipment are aboard. The round number of 5,000 pounds is used for their total weight. "Fully loaded" means that 10,000 pounds of load has been added to the light vehicle. In all cases the center of gravity of passengers or cargo has been taken as 92 inches from Station 0, the perpendicular at the extreme stern.

Version	Condition	Weight (pounds)	Trim (inches)
Propelled by Twin Outboard CP Propellers	Light	43,670	5.5 by bow
	Troops	48,670	8 by stern
	Fully loaded	53,670	19 by stern
Propelled by Tracks only	Light	41,990	25 by bow
	Troops	46,990	16 by bow
	Fully loaded	51,990	4 by bow

For interpreting degrees of trim to inches of trim, use a waterline length averaging 25 feet and find 1 degree equivalent to an average of 5.25 inches. To avoid confusion in the meaning of trim, the definitions in Section 2.0 Appendix A should be read.

The trim in the static condition is not the attitude of the LVTPX12 when under way at full speed. As remarked in the sections on model tests, the LVTPX12, whether propelled by screws or tracks, trims by the stern when under way, so that static bow trim actually produces the most favorable attitude when running. This is as true for the screw propelled vehicle as for the track propelled, but as shown in 4.4 an appreciable initial trim by the stern is an advantage when going at high speed, 10 MPH or more. This will be true especially in rough water.

4.9 Dynamic Analysis. The probable motions of the LVTPX12, particularly the periods of roll and pitch, are not without interest. Their exact calculation is impossible, for reasons to be shown, but their approximation can be close enough for the purpose of rendering broad judgment.

In roll, pitch, and heave, the craft may be likened to a spring-mass model in which the sum of kinetic and potential energies remains at all times a constant. This model ignores the damping losses from viscous friction and wave-making, but such losses are small enough to have only a little influence on the calculated natural period of motion. The important approximations are of the added mass or added moment of inertia due to the acceleration of water in the immediate neighborhood of the hull. Rational calculations of added mass for a number of simplified models, such as spheres, ellipsoids,

wedges, and boxes, are available, verified by experiment, but the complicated undercarriage of the LVTPX12 entrains an unknown mass of water during motion. That effect must be approximated, but because the natural periods of the vehicle, T_P for pitch or T_R for roll, are calculated by taking square roots, the total effect of all the estimates used in predicting the natural periods will not be far from right.

In Appendix A, Section 22.0, will be found the calculations for natural rolling period and natural pitching period, together with the assumptions made for added mass and added moment. The results are:

T_P = 2.1 seconds

T_R = 2.3 seconds.

Although the periods of pitching and rolling are very close together, this would be expected in a craft of such low length-beam ratio.

The significance of the natural periods of motion is in the response to the seaway. The excitation will be most severe when the period of encounter is very near the natural period of motion, that is where the value of the tuning factor is in the neighborhood of unity. For this condition of synchronism, the LVTPX12 will have to be in waves having regular length from crest to crest of 20 to 30 feet. The speed of such waves would be only about 6.5 knots, such as would be produced in a gentle breeze, when the speed of the craft with respect to the advancing crest were zero. When advancing at 10 MPH into waves of about 45 foot length, the vehicle will be in danger of synchronous pitching. This length corresponds to a breeze of approximately

12 to 15 knots. (Both length and height of waves depends on the fetch of wind, so that there is no exact relation between wind speed and wave length).

These conditions are not very severe. At other times, when the craft is in waves of greater length, both rolling and pitching periods will pretty well match the period of the waves. These predictions, of course, are based on the assumption of simple waves, which do not actually occur in nature. No matter how long the basic waves, there will always be some small ones to match the period of the vehicle. As shown, these will be short waves, of maximum height not more than 2 feet.

4.10 Summary of Research on Water Performance. During this research and design of a new amphibious assault vehicle for the U. S. Marine Corps, Chrysler Corporation has accepted the original specifications as a broad definition of an engineering task rather than simply as a limited set of requirements barely to be met. The specifications have not been regarded as boundaries on the resourcefulness of Chrysler engineers. Not only has Chrysler met the specifications on water speed, but Chrysler ingenuity has provided something better.

4.10.1 Water Speed. The minimum specified speed of 8 MPH with tracks has been met with a reasonable margin. By twin propellers, the Chrysler LVTPX12 will go in excess of 10 MPH. This vehicle especially would be most flexible in operation, for the operator could engage either tracks or screws, independently or together. This vehicle would make in excess of 6 MPH under tracks alone, even though it would not have the necessary bow fenders and stern appendages for maximum speed under track propulsion.

If the Marine Corps chose, however, the appendages could be included so that the LVTPX12 would go either at 8 MPH by tracks alone, or 10 MPH by propellers alone.

4.10.2 Backing Speed. Because the model used in the tests was not decked over, it could not be propelled backwards at more than 5 MPH for the prototype. The LVTPX12 will therefore back at more than 5 MPH. Although the specified backing speed was only 3.5 MPH, Chrysler has provided backing thrust far in excess of the specifications.

4.10.3 Maneuverability. The specifications call for a vehicle of maximum attainable maneuverability. The Chrysler LVTPX12 is not only maneuverable in executing turns, stopping, and backing, but moreover is steerable on a straight course to a degree never before attained in a tracked amphibian. The boat bow of the Chrysler LVTPX12 does not shed the large vortices which have hitherto been the chief reasons for the erratic steering of square, box-like amphibians. The water slides smoothly around the Chrysler LVTPX12, so that even the track propelled version will steer better than any previous amphibian. The high backing thrust will enable the Chrysler LVTPX12 to execute turns with unusual facility. The screw propelled Chrysler LVTPX12 when in the water will handle like a boat, the steering moment being applied by the driver through a pair of simple levers. Backing and stopping would be simply by changing the pitch of the propellers. The Chrysler LVTPX12 in the screw propelled version will turn around in its own length from a stopped position.

4.10.4 Stability. The Chrysler LVTPX12 is highly stable. A moment of 1,750 pounds-feet will be required to heel the vehicle 1 degree when it is floating. At a heeling angle of 90 degrees, the Chrysler LVTPX12 still has a positive righting arm of 1 foot when in the light condition. While the specifications call only for positive righting arm at 90 degrees, Chrysler designers have provided a liberal margin of safety.

4.10.5 Endurance. The track propelled Chrysler LVTPX12 will carry fuel for the specified 7 hours at 8 MPH in the water. The screw propelled version, however, will not require as much fuel during 7 hours of water travel as the amount specified for land endurance. Therefore the Chrysler LVTPX12, when propelled by twin outboard, controllable pitch propellers, will go over 13 hours in the water without refueling. Again Chrysler has exceeded the specifications.

4.10.6 Seaworthiness. The great range of positive stability of the Chrysler LVTPX12 provides insurance against capsizing in heavy surf. Watertight decks and hatches, automatic closers on all aspirating vents, and stout construction will enable the vehicle to cope with heavy seas. The maneuverability of the Chrysler LVTPX12, the ability to turn propellers and tracks both at once, and especially the ability to do this without lowering the power output of the engine, will enable the Chrysler LVTPX12 to maintain its attitude in 10 foot plunging surf and thus avoid broaching. The reserve buoyancy of both the track propelled and the screw propelled versions is 80 percent of displacement.

4.10.7 Launching. From the flooded well of an LSD or an LPD, the track-propelled Chrysler LVTPX12 will be highly maneuverable because of the large backing thrust provided in the Chrysler track design. If it is necessary to retract bow fenders and stern baffles, the vehicle will still retain ample maneuverability. The screw-propelled version will maneuver when tightly crowded, because the propellers will produce thrust even when housed under most conditions of loading and trim, or the driver may use tracks at his option. The screw-propelled vehicle will be particularly adept at boarding an LST in open water, because the tracks may be turned while the propellers are being used for steering.

4.10.8 Lying Alongside. When taking or discharging cargo alongside quay or ship, the Chrysler LVTPX12, either the track or the screw-propelled version, will not be vulnerable to damage. The propellers, when housed, do not protrude beyond the side of the vehicle. The propellers will operate in any position, and produce sufficient thrust for maneuverability even when housed. The tracks can always be relied on for movement.

SECTION 5.0
ANALYSIS OF ARRANGEMENTS

5.0 ANALYSIS OF ARRANGEMENTS

The arrangement plan of the LVTPXl2 cannot be divorced from the attainment of the primary aims in the performance of its chief functions. Hence, the arrangement plan cannot be consummated until the functions of the vehicle are defined in the order of their importance. Moreover, the effect of arrangement on this expected performance cannot be undertaken until all the facts are reviewed. When the accumulated facts from model tests, powering studies, structural analysis, track design, armor selection, and each aspect of this complicated military tool, are combined with the accepted primary aims, the arrangement is guided by the restrictions of logic.

In the attainment of desired performance, however, the relative merits of some arrangements are not manifest at a glance. While water speed, for example, might be accepted as a priority target, the decision whether to adopt mechanical transmission or electric drive cannot be made without exhaustive studies of each. To be certain that no efficient combination of components, spaces, and geometry has been overlooked, every promising idea deserves exploration to the point where its merits are conclusively demonstrated.

This analysis encompasses the following arrangements:

- Axiomatic requirements
- Specified features
- Water speed as a prime object
- Water speed and dual engines
- Water speed and a narrow vehicle
- Effect of a rear engine
- An air return track
- The configuration of jet-propelled craft
- Cycloidal propellers
- Electric Drive
- Water speed and single engine
- Selected concepts
- Details of concepts
- Alternatives

5.1 Axiomatic Rules for Arrangements. The two most indispensible components inside the vehicle and the components around which all other space must be arranged are the engine and transmission which must either be arranged in close proximity or separated by some distance. It would be possible to place dual engines on either side of the vehicle anywhere within its length; however, if dual engines were mounted together, either in parallel or in tandem, they would amount to a single engine. Four possible basic arrangements exist with a single engine:

- Engine forward, transmission forward
- Engine aft, transmission aft
- Engine forward, transmission aft
- Engine aft, transmission forward

Without extenuation, the following characteristics of the LVTPX12 will be accepted as axiomatically proper, in this order:

- No water speed less than 8 MPH with tracks or 10 MPH with auxiliary devices at maximum possible efficiency will be accepted.
- No arrangement which requires reduction in armor protection below specified values will be worth consideration.
- The aim in arrangement shall be to stay within a length of 26 feet, a width of 10.5 feet, and a deck height of 8.5 feet, and to reduce any or all of these if possible.
- Those arrangements resulting in minimum weight will be entertained with favor.
- The LVTPX12 is expected to be an assault vehicle. Therefore, the effectiveness of the gunner, his efficiency, vision, freedom of movement, will be values of a high order.

5.2 Specified Conditions Affecting Arrangements. The performance of the detailed functions of the LVTPX12 requires that certain spaces and facilities be provided. While meeting the axiomatic imperatives, the LVTPX12 must contain the specified spaces fitted into its geometrical limits, even before the satisfaction of any prime characteristic is attempted.

5.2.1 Exterior Conditions. Increasing the length would not only increase water speed but would also provide more usable inside space. However, transportation and maneuverability set a limit on length. Height is a tactical penalty, though a reward in water performance. Since neither length nor height are incompatible with other specifications, the maximum length of 26 feet and a maximum height of 8.5 feet will be accepted as the outside boundaries.

5.2.2 Interior Conditions. The length of the cargo compartment is specified to be at least 14 feet and the width to be at least 6 feet. The specified seating capacity is 25 men with field equipment. Although these two specifications are not altogether incompatible, seating 25 men in a space 6 by 14 feet (See Appendix N) cannot be done without objectionable results. The inside seating width must therefore be more than 6 feet.

This does not mean, however, that the LVTPX12 must necessarily be the specified 10.5 feet wide. The compartment length has important effects on other areas not directly related to arrangements: exterior shape, structure, and distribution of weights. The inside length must begin at the inside of the ramp. If the ramp, whether forward or aft, is raked, a part of the vehicle's length is not available for cargo length. If the ramp is plumb or nearly so, easy opening requires its

center of gravity to be outside a vertical plane through the hinge. Thus under any circumstances the cargo compartment must take up more than 14 feet of the vehicle's overall length.

5.3 Water Speed. If the decision is to accept maximum water speed as the overriding aim, then the mandatory characteristics may be taken from Section 4.2 on model tests. For maximum efficient water speed, the necessary characteristics most affecting arrangement are hull form and power.

5.3.1 Effect of Hull Form on Arrangements. The attainment of high water speed at best propulsive efficiency requires a fine bow, the finest that can be designed within the allowable length and the unavoidable displacement. Although the space within such a bow is largely usable, the fine bow leaves little option in location of ramp. It is not impossible to design a forward ramp to fit a boat bow but the structural difficulties, the operating complexity, and the geometrical compromises are so manifest that no detailed examination is necessary. At the best, some freedom would be lost in shaping the bow. It can be stated categorically that reaching maximum water speed with a forward ramp would be difficult.

The stern ramp is justified by additional advantages in performing the primary functions of the LVTPX12. Although no numerical proofs can be given for assertions on the tactical advantages of the stern ramp, conclusions from reasonable assumptions and from competent testimony are proper enough.

Some experienced officers of the Marine Corps have expressed the opinion that assault troops will be more confident when disembarking from a stern opening, where they have steel plate and a machine gun between themselves and the enemy, than from an opening where they emerge directly into the face of fire with the gun behind them. If the vehicle is turned around to put the bow opening away from the enemy, the gun is at the disadvantage of having to fire over the vehicle. In this position the gun cannot reach ground targets at close range and in fact when on the upward slope of a beach could not reach ground targets at any range. (See Gun Map Coverage, Figure 16-15, Section 16.0).

The awkwardness of turning the bow away from fire can be pictured especially in a rescue operation, where the rescue party must recover casualties and load litters under fire. To open the ramp facing the enemy in this situation is impractical. The gunner, firing backwards over his vehicle, cannot lay down an effective protective barrage.

The stern ramp would be a handy facility in laying communication lines. With wires paying out astern, the LVTPX12 could quickly hook up command posts over terrain impassible to other vehicles and impassible even to foot signalmen, all the while protected by its own armor and its weapons.

The same can be said for laying down concertina wire in a defensive situation.

For fast deployment in difficult terrain, the vehicle with the stern ramp need not stop and lower the ramp every time a man is unloaded there is no reason why the vehicle cannot move along with the ramp open.

The bow ramp does not suggest any special operational or tactical advantages for itself. In some configurations, the bow ramp would allow more favorable

trim. Also it is true that putting the engine in the stern removes noise, vibration, and heat from the driver's compartment, and that the exhaust pipe can eject directly aft. These, however, are all engineering problems, not operational problems, and any one of them is not of large significance by itself. Since the conditions for maximum water speed and tactical utility cannot be changed, it is better to adopt the stern ramp for these operational advantages, and to apply the designer's resourcefulness to the minor difficulties.

5.3.2 Effect of Water Speed on Engine Arrangement. Whether the LVTPX12 is propelled by tracks or by other devices, top water speed will require 800 horsepower installed. It has been shown in Section 4.0 that this is the desirable power even with screw propulsion. The fine bow and the stern ramp mean that a single engine cannot be in the stern. This will be true no matter what the means of auxiliary propulsion. Dual engines can be on the track sponsons, but if located aft they must not reduce the inside clearance in the cargo area to less than 6 feet.

5.3.3 Effect on Armament and Armor Protection. There is nothing about the location of an engine forward to interfere with efficiency of weaponry. In fact, the gunner's fixtures in this arrangement are permanently installed in the most favorable location. Nothing need be cleared for the passage of cargo or passengers. The engine, enclosed in an individual box, does not interfere with the gunner's functions. The boat bow increases the degree of gun depression except over the very point of the bow.

The armor configuration imposed by the boat bow design provides additional protection for the gunner and crew due to the obliquities of form. This is particularly advantageous at the front of the vehicle from obvious tactical considerations. The magnitude and effects of this added protection are discussed in Section 6.0.

5.3.4 Summary of Criteria for Major Component Arrangement. The decision to attain maximum water speed in both configurations of the LVTPX12 has automatically established several fundamental characteristics. The boat bow is imperative, and engine power installed must be at the 800 level. The practical aspects of general arrangement lead to adoption of the stern ramp, which imposes additional restraints on vehicle design. A number of variations of possible designs suggest themselves and the advantages and disadvantages of each will be evaluated in the following paragraphs.

5.4 A Dual Engine Arrangement. If maximum water speed, along with the axiomatic and specified characteristics, are selected as the primary aims to be satisfied by arrangement, there is nothing about two engines that would defeat the aims, except the availability of engines powerful enough, narrow enough, and light enough to do the job.

Schematically, this arrangement would appear as follows. The engines would not necessarily be at the extreme stern.

5.4.1 Advantages and Disadvantages of Dual Engines. Not all the apparent merits of a dual engine arrangement are definitely provable.

- Dual engines, installed atop the track sponsons, could be anywhere along the length, thus allowing any chosen trim. This is true if the gross vehicle weight is to remain the same at all times, or if cargo, passengers, and fuel can always be loaded with center of gravity at the craft's center of flotation. If loads are not at the center of flotation, which is on the average 144 inches from Station 0, the LVTPX12 will undergo a change of trim. When 10,000 pounds is loaded at 92 inches from Station 0, for example, the change of trim is 18 inches.
- Dual engines provide redundancy and therefore greater reliability despite the fact that a failure is twice as likely to occur. This is true because, if one of the pair fails, sufficient power for travel is still supplied to the single transmission. However, the complexity of the installation is far greater, as can be observed from

the layouts to be shown, and therefore the probabilities of failure are increased accordingly. Any assertion that dual engines would definitely provide more safety than a single engine is subject to challenge.

- Dual engines allow wider choice between manufacturers. True only if water speed or weight are to be sacrificed. If flexibility of location is to be preserved, the width of dual engines must not cause them to reduce width of cargo space to less than 6 feet. In Section 8.0, the GM 6-71T is shown to be the only engine with the width and power to do the job. The Allis-Chalmers Model 3500 engine is narrow enough, but is rated at only 230 HP. The LVTPX12 with the pair of Allis-Chalmers 3500 engines will weigh the same as with the single GM 12V71T, but will not go as fast. Although a pair of GM 6-71 engines will develop 800 horsepower, frictional losses will result in less speed than with the single 12V71T, and the vehicle will weigh more. (See Sections 5.2 and 5.3 in Appendix D).

- By allowing a lower floor, the dual engines would allow a lower silhouette. Not true. The dimensions on the sketches in Appendix N show that the engines themselves require a fixed height between track sponson and top deck, and the only way to lower the deck is to lower the track sponson. This can be done, of course, by changing suspension geometry resulting in poorer ride characteristics.

5.4.2 <u>Detailed Arrangement of Dual Engines</u>. Figures 5-1, 5-2, and 5-3 show the fairly complete layout of an LVTPX12 with two Allis-Chalmers 3500 diesel engines. The stern ramp and the boat bow conform to the best hull for efficient propulsion. This vehicle can be propelled either by tracks alone or by twin outboard screws.

In most respects the curves of hydrostatic characteristics in Section 4.0 would apply to this vehicle, because the hull is the same as that for which these curves were calculated. Curves having to do with metacentric height would be shifted slightly, however, because of the greater height of vertical center of gravity. No detailed calculations of centers has been made for this distribution of weights.

The track and suspension system is virtually the same as for the vehicle with single engine, except that the forward transmission requires a front drive.

The two Allis-Chalmers 3500 engines produce a total of 460 horsepower, sufficient in all respects for land performance. The expected water speed by tracks would be 5.9 MPH. By twin screws, these two engines would drive the LVTPX12 at 9.4 MPH. The primary reason for the Allis-Chalmers engines being in this layout is their light weight and narrowness. Whereas the Allis-Chalmers 3500 engine requires a width of only 19 inches, the GM 6-71T requires 25 inches. Each AC 3500 weighs 1,400 pounds, while the GM 6-71T weighs 1,802 pounds. Thus if the GM engines were adopted, the available inside clearance would be reduced by 12 inches, and the weight would be increased by 800 pounds. Although the GM 6-71T engine could be fitted into

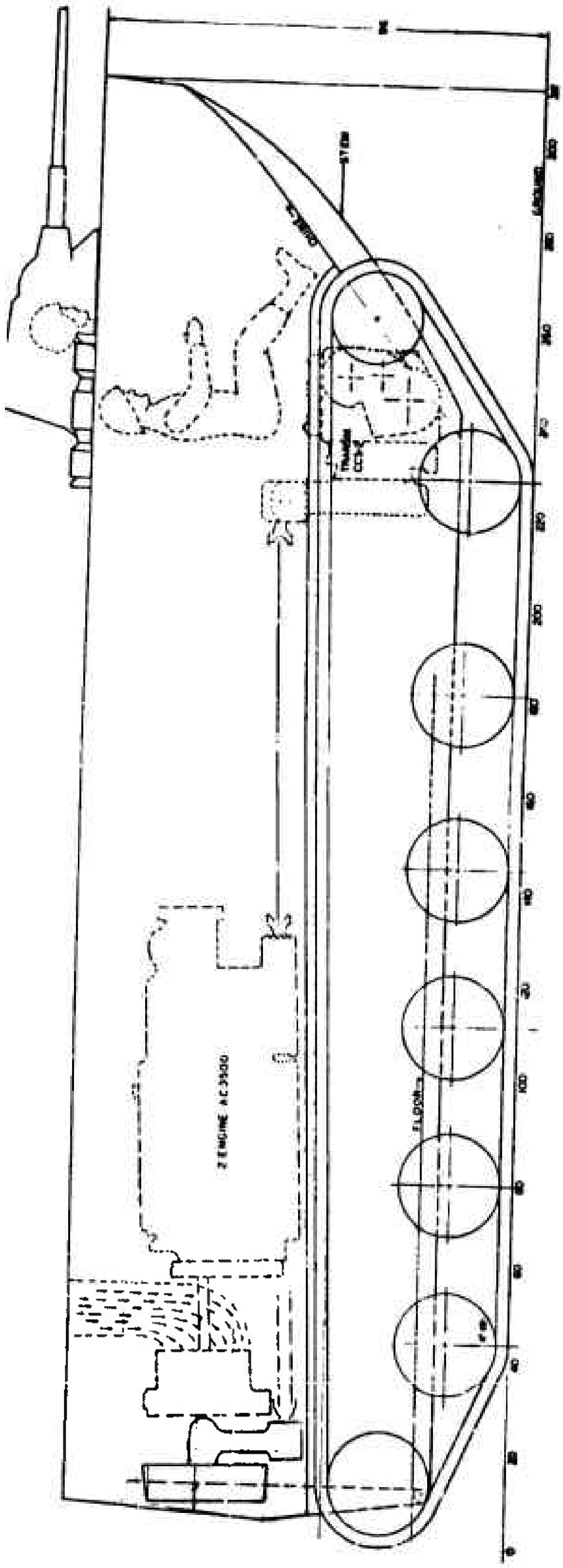

Figure 5-1 Dual Engines Arrangement - Elevation View

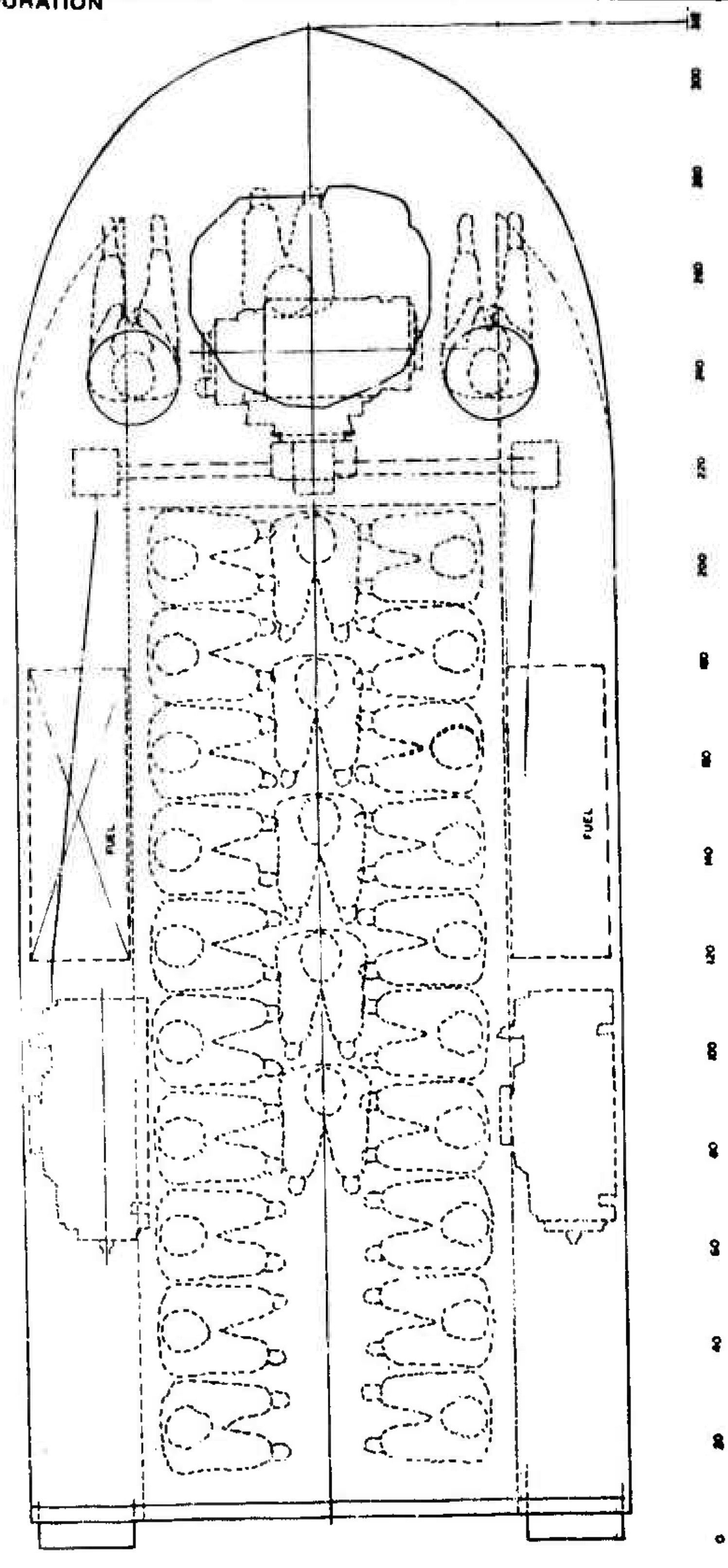

Figure 5-2 Dual Engines Arrangement — Plan View

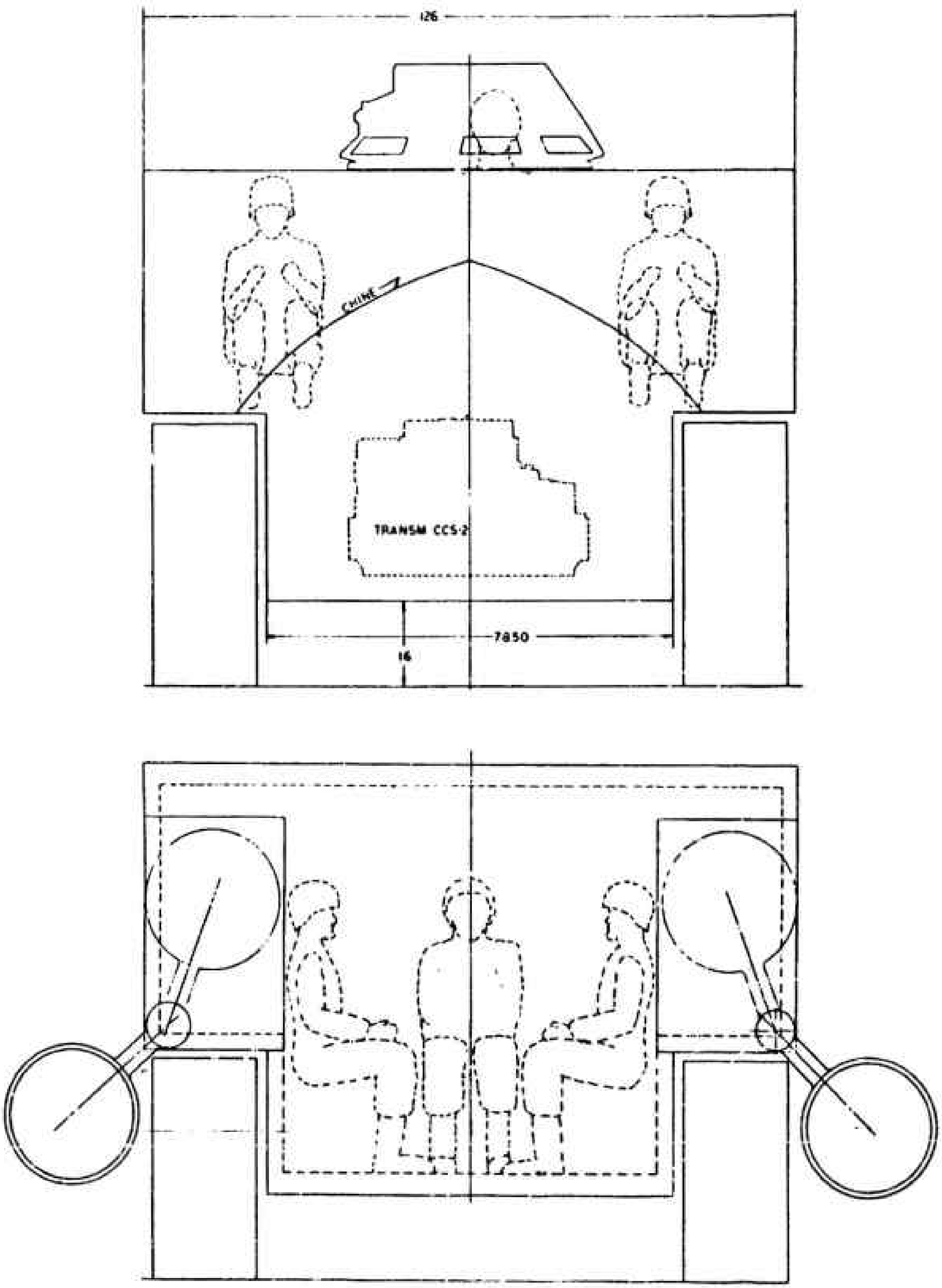

Figure 5-3 Dual Engines Arrangement - End Views

the available height, access to the top of the engine would have to be from outside through a hatch, or else the engine would have to be removed. The injectors -- the parts of a diesel engine most frequently requiring attention-- are in the top of the engine.

An adequate number of seats for passengers can be provided with this arrangement. Some of them will have to face rearward, or else some superfluous motions will occur during disembarkation. On each side, the heads of two passengers will be within a few inches of an engine. A detailed study of the ventilation of passenger space in this arrangement has not been made. Water cooled exhaust manifolds would increase the width of the engines and would require approximately 50 percent more radiator capacity. With dry manifolds, the air supply for the engine can come first across the manifolds if the engine is enclosed in a box, but the box must clear the engine enough to allow passage of the air without choking. Such enclosures around these side engines will invade the cargo and passenger space. Thus,the ventilation and cooling of engine and passenger space in this arrangement present added design problems.

5.4.3 Other Features of the Dual Engine Installation. As in the case of the single engine, some points are good, some are bad:

- Drive sprocket location. Front. It is shown in Section 9.0 that this is less desirable than a rear drive. But successful tracked amphibians have been built with front drive.
- Ramp. Can still remain in the rear
- Bow Form. Freedom not affected.

- Vertical center of Gravity. High. Although no detailed calculations have been performed, a positive righting arm would probably remain at 90 degrees of heel.
- Space for Crew. Ample.
- Accessibility of machinery. Transmission accessible at all times. Engine not accessible under certain conditions of cargo loading. Outside of engine not accessible at any time.

- Relative weight. Weight increased by engines, drive system, and accessories.
- Space utilization. Generally good. Transfer cases would be required to keep drive shafts from blocking space above sponsons, but very little of this space could be made usable.
- Location of cooling. On each side, towards stern. Radiator wells would be symmetrical.
- Watertight compartmentation. Very difficult.
- Controls. Complicated.
- Structural problems. Special hatches required over engines. Turret must be built into a hatch, or else transmission must be moved horizontally back to cargo hatch for removal.
- Drive to propellers. Direct.
- Interior communication. Direct between members of crew and between crew and passengers.
- Ventilation of passenger space. Satisfactory. Exhaust directed aft, behind passengers.

- Engine noise. Engine removed from driver, behind him, but on each side of troops.
- Multisource engines. Limited, unless sacrifices are made in power.
- Weight concentration and trim. Favorable.
- Fuel location. Could be on top of sponsons, or under floor.
- Removal of machinery. Engines through hatches. Internal facilities required for transmission.
- Logistic support. More complex due to higher complexity of installation and more replaceable parts.

5.4.4 Conclusions on Dual Engines. Only one decided advantage obtains from dual engines; trim may be controlled more freely. A few miscellaneous advantages are canceled by some decided disadvantages.

The two engines either would increase the weight of the LVTPX12, or supply insufficient power. They would increase the overall acquisition and operating costs over the life of the program. As for water speed, the best the dual engines could do would be to meet the speed of the single engine.

For these reasons, the dual engine installation is less desirable than a single engine.

5.5 Maximum Water Speed with a Narrow LVTPX12. In Section 4.0 it is found that the resistance of a wooden model can be reduced some 8 percent by narrowing the beam. Although it is not certain that this same reduction would have prevailed had the tracks been simulated in detail, and not certain that the propulsive efficiency could be maintained, the narrow craft would still have

some advantages in the water. Its stability on course should be slightly better. It should make better speed in head seas. Its rolling period would be longer. Its ability to go between obstacles on land would be increased.

5.5.1 Arrangement of a Narrow LVTPX12. The general configuration is shown in Figure 5-4 and Figure 5-5. Figure 5-6 is a narrow vehicle lines drawing. The beam of this vehicle would be only 104 inches. To provide an interior cargo width of at least 6 feet, the track sponsons would be omitted in most of the length. With the exception of beam and the design of the track and suspension system, the narrow vehicle is, in virtually all respects, the same as the wide one. The reduction of beam makes the housing of side propellers impossible, however, so that if this narrow vehicle were propelled by means other than tracks, adequate inside space would become a serious difficulty. Installation of jet pumps large enough to produce 10 MPH would be no easier than installation of propellers. In fact, the best solution to the problem of auxiliary propulsion would be a single outboard stern propeller. This would eliminate the stern ramp and the boat bow. Consequently 10 MPH on water would be difficult to attain, especially because of the height of the bow wave created with a blunt bow.

The studies that have been made on the narrow LVTPX12 have not included twin outboard propellers. Attention has been concentrated on designing a track propelled vehicle with good performance both on water and on land. Calculations of the static characteristics of a narrow vehicle with 104-inch beam are found in Appendix A. Due to the increased weight of track and suspension system, the total weight of this design is not significantly reduced below that of the wide vehicle.

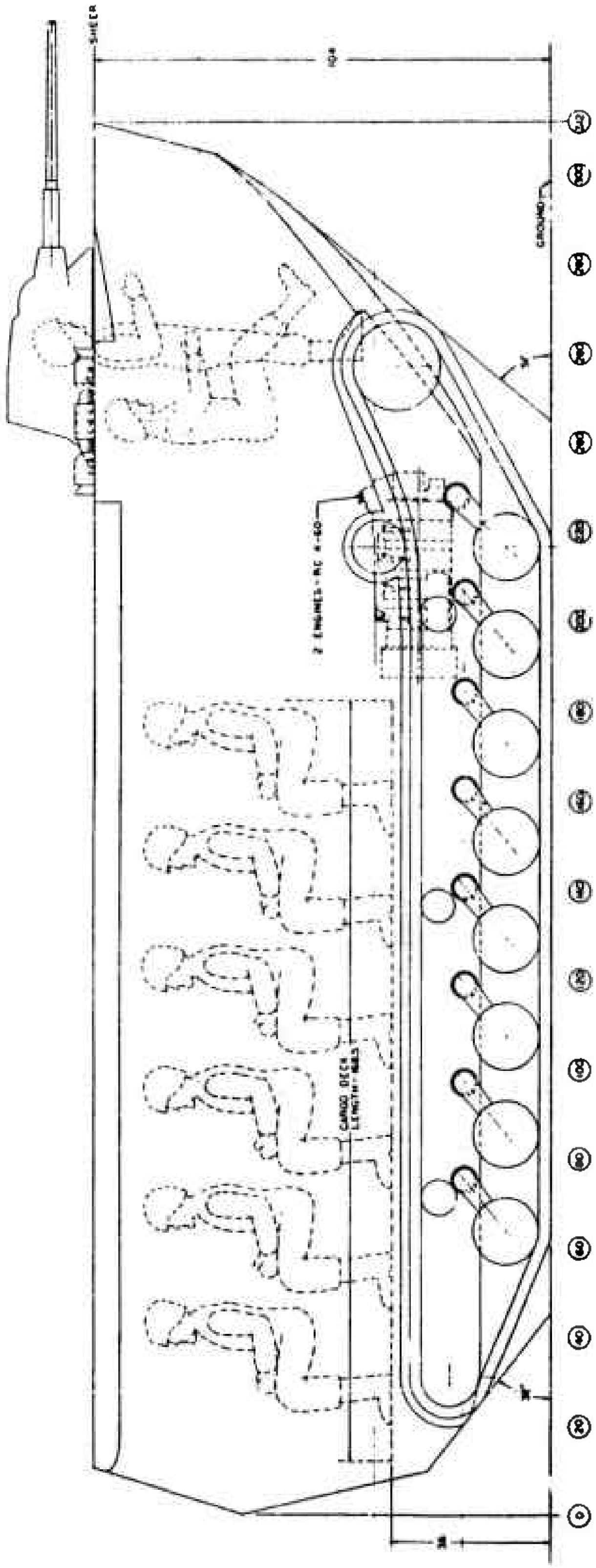

Figure 5-4 Narrow Track-Propelled Vehicle - Elevation View

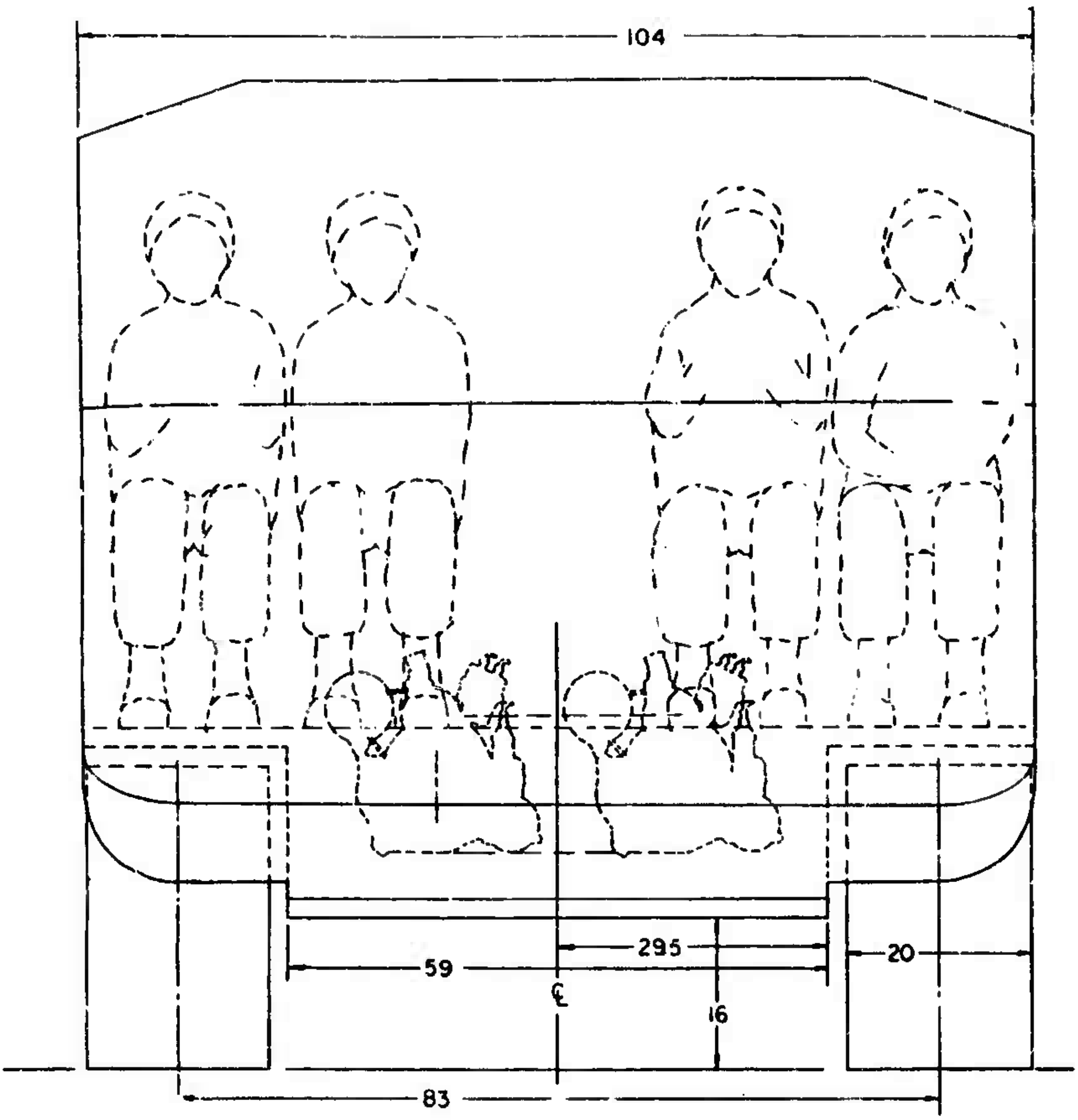

Figure 5-5 Narrow Track-Propelled Vehicle - End View

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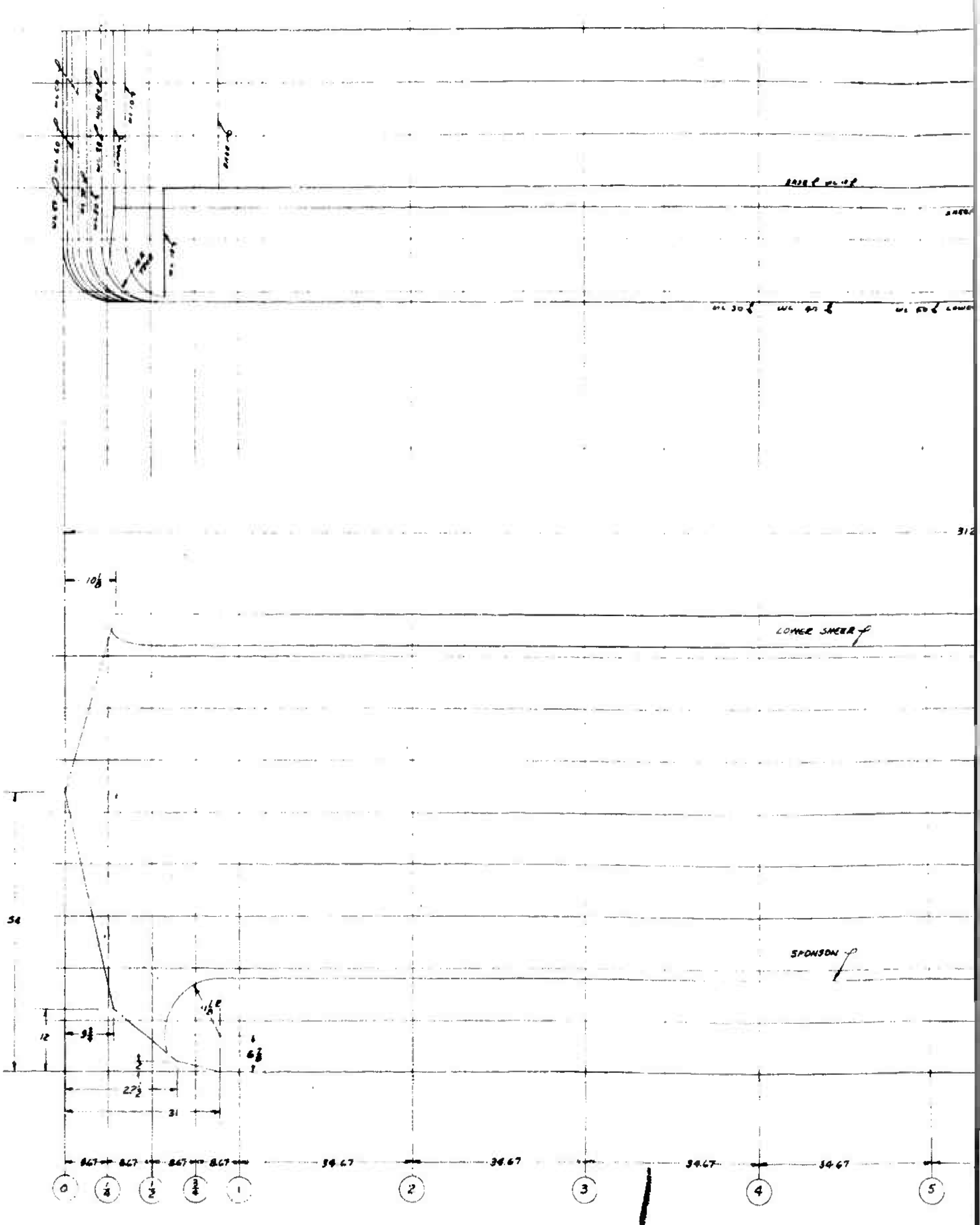

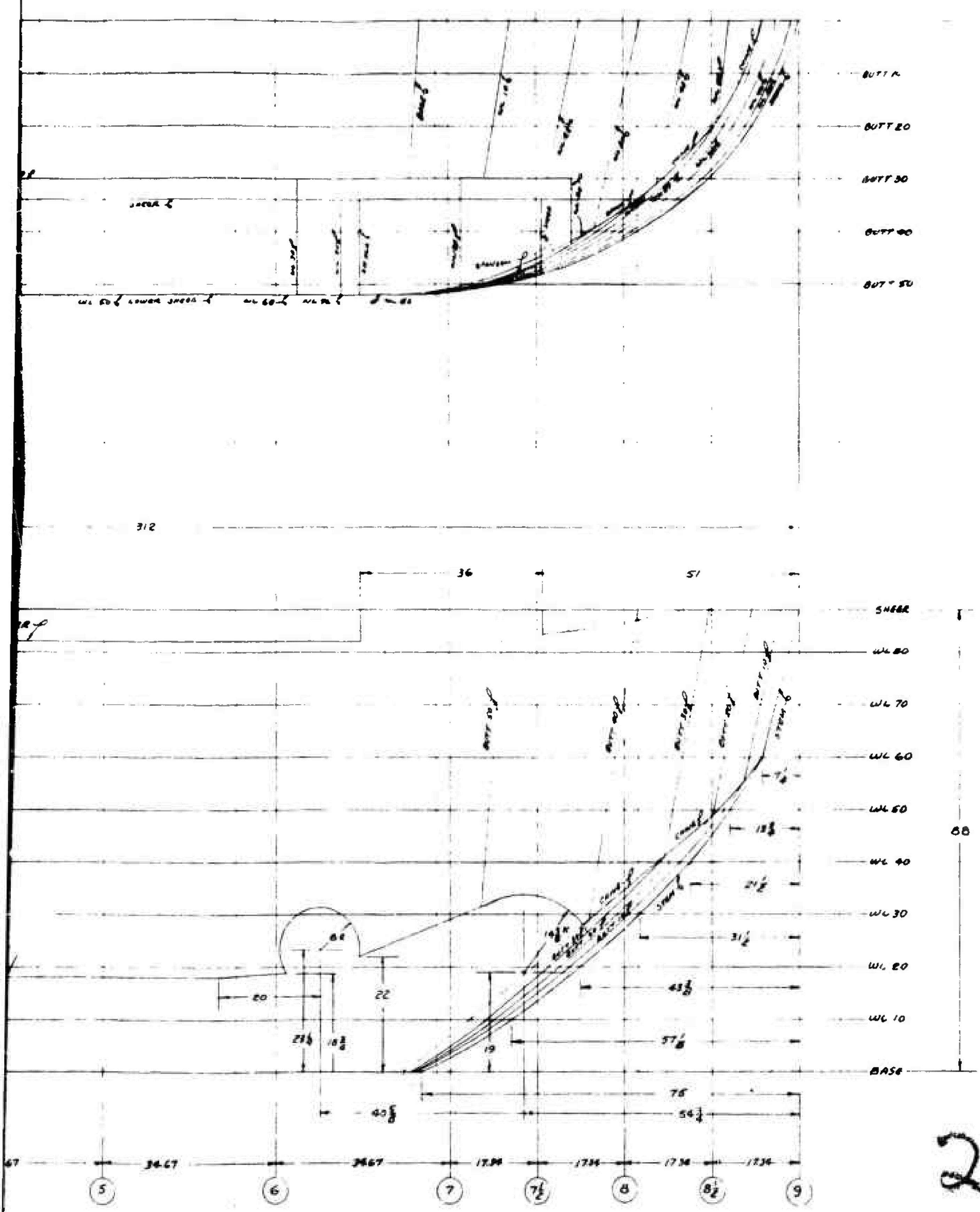

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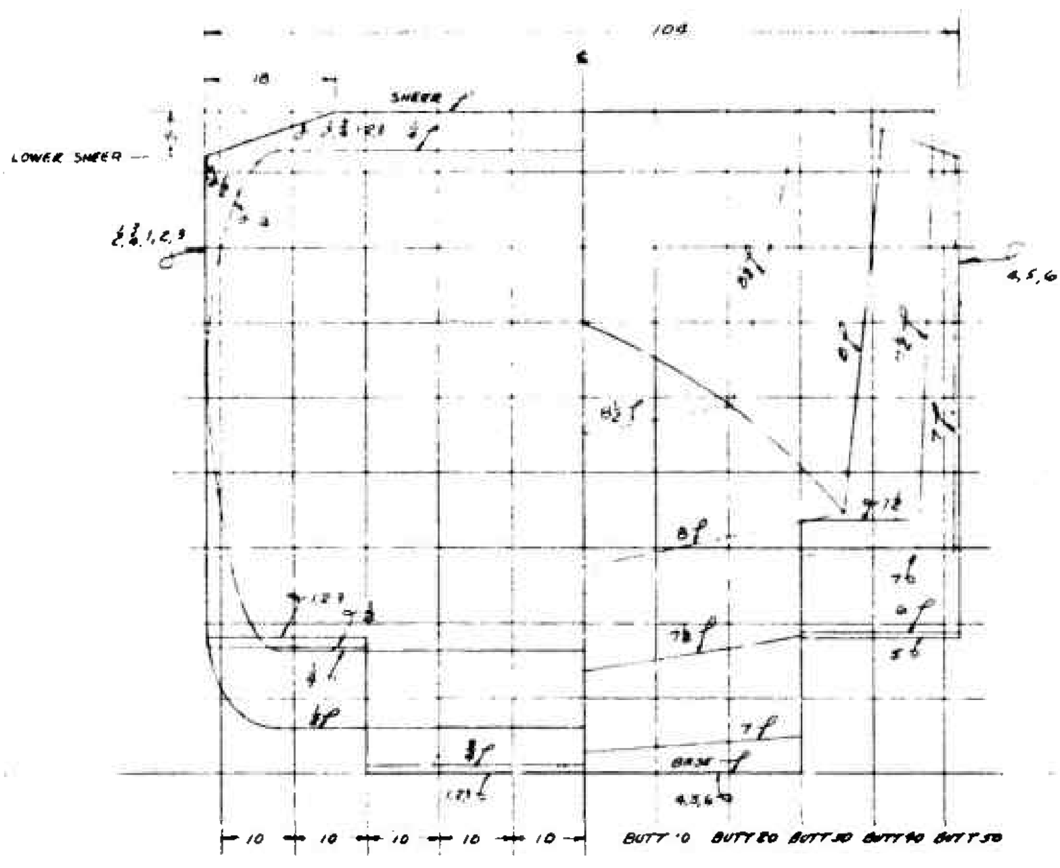


FIGURE 5-6 NARROW TRACK PROPELLED VEHICLE- LINES DRAWING

5.5.2 Design of Tracks for a Narrow Vehicle. In order to clear obstacles 3 feet high, the axle of the forward idler must be 36 inches from the ground line (Figure 5-4). This, plus the radius of the idler and the thickness of the track, puts the top of the track over the front idler 52 inches off the ground line. But at the same time, the narrowness of this vehicle requires the cargo deck to be flat all the way across the vehicle, if the minimum of 6 feet inside clearance is to be provided. In way of the cargo compartment, therefore, the top of the track must be less than 52 inches off the ground. In fact, the ground clearance, 16 inches, plus the 16 inches necessary for the transmission under the floor, provide a cargo floor height of 32 inches.

Thus the track must have a reverse idler abaft the forward idler. Crowding the entire track and suspension system of this vehicle into a vertical distance of only 32 inches is no simple exercise.

The low track envelope immedia equires a succession of compromises. To obtain a tolerable wheel travel, the road wheel diameter has to be reduced from 20 inches to 15 inches. The torsilastic suspension, found to be preferable in the analysis of Section 9.0. cannot be adopted because there is no room to mount the rubber springs outside the hull and still maintain an adequate number and proper spacing of road wheels. The alternative is torsion bar suspension.

The small diameter of road wheels reduces the amount of contact surface with the tracks and requires an increase in number of wheels. To obtain adequate track guidance the number of wheels must be increased to eight per side. The rear drive sprocket must be reduced in diameter, therefore the track

pitch must be reduced. This immediately adds to the number of links, pins, seals, and bushings.

The wheel travel in the resulting design is only 9 inches, as compared with 12 inches, the desirable distance. The torsion bar suspension requires a shock absorbing system. The resulting design can be seen in the illustration to be rather tightly packed.

The following observations can be made about this design:

- The road wheel jounce of only 9 inches will cause severe pitching in cross country travel and will limit the speed to 10 MPH under such conditions.
- The track and suspension system weighs 1200 pounds more than the design for the full width vehicle.
- The number of road wheels, arm assemblies, track shoes, pins, and idlers has been increased in cost and complexity. Maintenance has been increased. Inventory of spares has been increased.
- The shock absorbing system increases first cost and maintenance. It decreases the reliability of the vehicle.
- Clearances between moving components are so small that the probability of fouling is increased.
- The small road wheels increase rolling resistance and will have poor tire and bearing life.
- The steering ratio (L/t) is 1.8, leading to poor handling of the vehicle on land and large power consumption in maneuvering.

- The restraining rollers atop the track produce a reverse bend in the track and add to the complexity and to the weight. They increase the opportunity for fouling and decrease track seal and bushing life.

5.5.3 Conclusions on a Narrow Vehicle. For any given power, no significant improvement in water speed can be obtained with reduced beam. The only feasible auxiliary propulsive device for such a vehicle would be the single stern propeller whose consequences are surveyed in Section 4.0. The low clearance for the suspension of the narrow vehicle, although workable, produces an undercarriage of mediocre quality. The suspension is a vital part of the system. Therefore, reduction in beam is not worthwhile.

5.6 A Rear Engine as a Desirable Feature. The number of possible arrangements in an amphibian is large. If water speed is not selected as a primary aim, other possibilities immediately open themselves to review. Suppose for example that a rear engine is chosen as the most desirable arrangement.

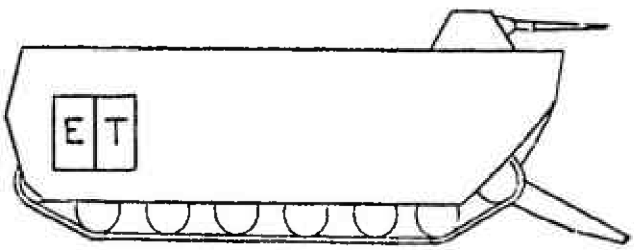
5.6.1 A Forward Ramp. Engine in rear obviously means a forward ramp. Several desirable results might be obtained if it were decided that the boat bow were not worth retaining, and that the inherent operational virtues of the stern ramp could be foregone. With ramp forward and power plant aft, the craft would be down by the stern, or at least trimmed by the bow only a very little, under all conditions. While it might be contended that this stern trim in the static condition would be desirable, the LVTPX12 would trim by the stern severely when under way at high speed --- assuming that high speed could be attained. If it seems that this should be no more objectionable on the LVTPX12 than on the LVTP5, the LVTPX12 is shorter than the LVTP5 and

hence more sensitive to weight distribution. Any kind of auxiliary propulsive device, of course, would result in an intolerable amount of weight in the stern.

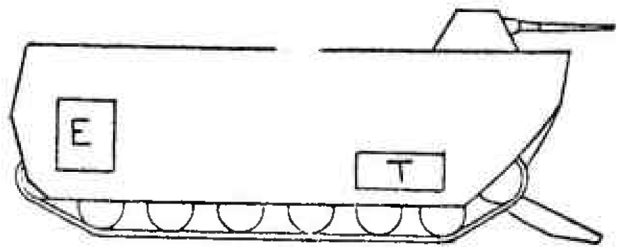
5.6.2 Effect on Spaces. With a forward ramp, the hull can be subdivided with a watertight bulkhead forward of the engine compartment. The engine thus exhausts out the stern and its noise is bulkheaded off from passengers and crew.

A special hatch for engine removal must be provided. The space above the engine becomes useless in the operation of the vehicle, but can be used for stowage of various items that add still more weight in the stern.

Only two arrangements of engine and transmission are possible with the forward ramp:



- Engine aft, transmission aft. In the light condition, the static trim by the stern would be extreme, and the additional dynamic trim would prevent the attainment of high speeds. Auxiliary propulsion of any kind in the stern could not be tolerated with this arrangement, because of the excessive concentration of weight.



- Engine aft, transmission forward. This arrangment would not concentrate so much weight in the stern. A front drive sprocket would result. In all other respects, this configuration would remain the same as with engine and transmission aft.

An arrangement with engine aft and ramp forward would make the single propeller the most eligible candidate for auxiliary propulsion, at least from the standpoint of weight and steering. In Section 4.0, however, it is shown that the single propeller is much less efficient than propellers on the side. The remarks in Section 4.0 on vulnerability of the stern screw, trimming and heeling moments, and on the high thrust deduction resulting from the hull in front of the propeller, should be recalled.

Disadvantages of the front ramp arrangement, other than the previously discussed operational problems, include the undesirable location of the gunner in the cargo space. This location interferes with troops embarking and debarking through the ramp and significantly decreases the effective stowage area when transporting cargo during combat operations. The decrease in angularity of the bow plates likewise provides less armor protection per pound of plate in the vital bow area compared with that provided by the boat bow.

The sum of these evaluations is that the objections to the rear engine and the bow ramp outweigh the few advantages.

5.7 The Effect of an Air Return Track on Arrangement. The model tests on the LVTPX12 (Section 4.0) do not show conclusively that overhead return tracks either would or would not improve the propulsive efficiency in water. The possible benefit, however, is shown to be small. Nevertheless, supposing that tracks returning in air would be desirable, the effects on arrangement are examined.

5.7.1 Bow Shape and Air Return Tracks. When the tracks are under sponsons, the protrusion of the front sprocket beyond the side produces a measurable resistance, even though a large part of the water flows around the bow above them. If tracks return over the hull, the design of a boat bow can readily be seen to be pointless. The only way to obtain fineness is to design a scow bow, such as those on the earliest tracked amphibians. Although this is better than no fineness at all, it is not as good as the boat shape, especially in waves. The scow bow can easily accommodate a forward ramp. The inside space above the ramp hinge line, however, becomes useless. For example, if the ramp were 6 feet wide, 5 feet high, and raked 45 degrees, 75 cubic feet above the ramp hinge line would be unusable. This also would mean that 5 feet of the inside length would be lost, since the cargo space and engine space must be measured at the floor, not the top deck. The stern ramp could be retained, of course, with the overhead return track, and could be plumb so as to avoid this loss of interior space.

5.7.2 Effect of Air Return Tracks on Structure. If the tracks returning over the deck were not covered, they would be dangerous to crewmen in mooring, towing, or any other duty requiring a man on the weather deck. Hence, a recess must be provided in the hull from end to end. The consequence of this will be added framing and added weight. All vents and hatches must be inside the tracks.

Design of wells for retractable propellers would not be difficult with the air return track, but the wells could not be left open abaft the screws, because the tracks would be in the way. The thrust of propellers when housed would therefore be diminished. (See Section 24.0 of Appendix A.)

Although no detailed study of weights and centers has been made of the LVTPX12 with overhead return tracks, the vertical center of gravity would obviously be raised. The recesses for the tracks in the top deck would cause a loss of righting arm in heeling and thus diminish the range of stability.

5.7.3 Weight, Cost, and Complexity of Air Return Tracks. Wrapping the track entirely around the hull would require it to be some 10 feet longer. Together with the added number of rollers and idlers, the weight would increase by about 1,600 pounds. The added length of track may be translated into the extra number of track pads and pins. The production cost of the vehicle would increase significantly. Because the upper rollers could not be the same as any of the wheels or idlers in the suspension system (too big and too heavy), a number of parts would be added to the system. Replacements would have to be carried for these.

An intangible factor, but not an insignificant one, would be the noise generated by tracks rolling overhead, especially at the speeds the LVTPX12 is expected to travel.

5.7.4 Conclusions on Air Return Tracks. The only justification for the adoption of overhead return tracks would be the expectation of higher water speed. Although the model tests do not prove definitely that the air return track is inferior at high speed to the track returning under the sponson, the tests do show that the probability of superiority is not very good. The chances of any improvement with the air return track are diminished still further by the necessity of a scow or barge bow instead of the boat bow. The most likely outcome would be a decrease in propulsive efficiency rather than an increase.

In addition, the overhead track is characterized by a number of difficulties, increased cost, and increased weight. This configuration is therefore not worthy of further consideration.

5.8 The LVTPX12 Propelled by Hydrojets. Propulsion by hydrojets is examined exhaustively in Sections 6.0, 7.0, and 8.0 of Appendix A. The subject here is not the hydrodynamics of jet propulsion, but rather the effect on arrangement that would ensue if jet propulsion were selected as a desirable characteristic, and the consequences of this choice in the operation of the LVTPX12.

5.8.1 A Side Installation Over Track Sponson. Although 26 inches of width on each side are available for jet pumps, the problem here is to get a supply of water for the pumps. If the intake is directly upward the water must come through the track and suspension system, so that the loss would be high. If the intake is in the side of the vehicle, air will be admitted under some

conditions of trim. In a seaway, air will be admitted as the vehicle rolls. Figure 5-7 shows a swiveling intake. For operation of the jets, these intakes swing outboard on each side and face downwards to put the opening as far below the surface as possible. To provide maximum intake area, the opening is made oblong instead of round. The entire intake duct swings around the drive shaft as an axis. The maximum diameter possible with this arrangement is limited not by the width available for the impeller so much as the area allowable for the intake, since there is no point in having an impeller larger than the intake.

The hydraulic losses in this system would be higher than for a bottom installation, because of the extra bends in the duct. The oblong shape of the inlet is against hydraulic efficiency. The drag of the intake must be taken into account. In severe rolling, this intake would still admit air once in a while. The entrained air near the surface of the water alongside the craft at high speeds would always be taken in. This will promote cavitation.

No calculations of the probable propulsive efficiency of this system have been made. The efficiency would be less than that of the bottom installation examined in paragraph 5.8.2. While impellers somewhat larger than 18 inches can be admitted, the one designed here is 18 incnes, because, if the duct diameter is increased, the depth of immersion when the duct is outboard has to decrease. While the efficiency of the intake can be improved by making it wider, the resistance of the submerged duct will thereby be increased.

This installation would not operate with the ducts housed.

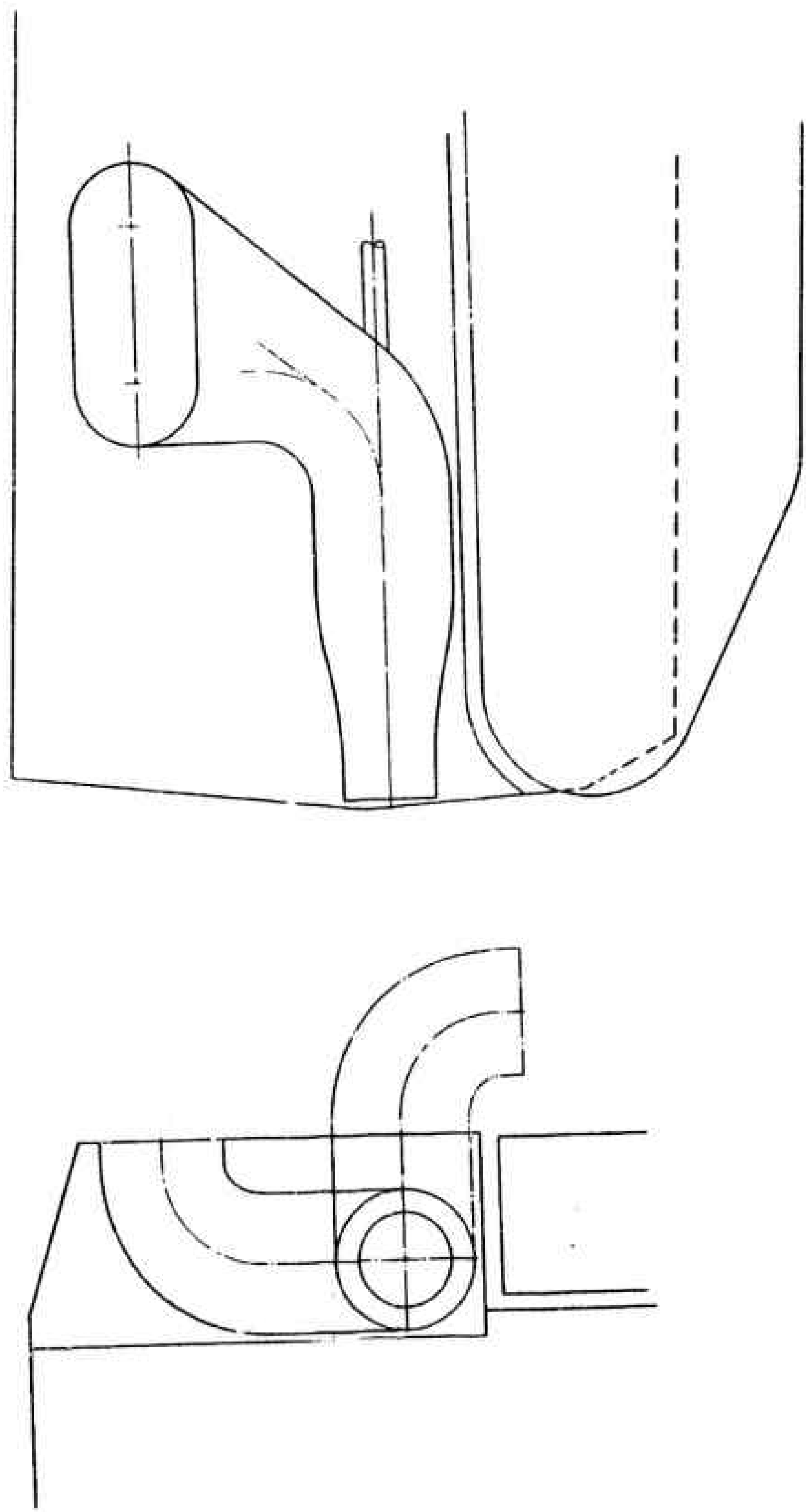


Figure 5-7 Intake Duct for Side Jet

Arrangement of interior spaces and machinery would be no special problem with these side jet ducts. Shafts could be led from a forward single engine, or the dual engine arrangement could be adopted. The ducts need not reduce the width of cargo space. Passengers would be seated in the manner illustrated in Figure 5-2.

5.8.2 Hydrojets in the Bottom. Figure 5-8 shows a study for two ducts and 18-inch impellers in the bottom. A detailed analysis by calculation for the performance of this installation is found in Section 8.0 of Appendix A. The design of this sytem began first with the largest impellers that could be nstalled in the bottom. Although the illustrations (Figures 5-8, 5-9, 5-10 and 5-11) do not show the boat bow and the plumb transom, the performance of the jets with this optimum hull shape can be calculated from information in Section 4.0. With a propulsive efficiency of 30 percent and 214 effective horsepower, the power input to the pumps would have to be 715 horses. (This prediction is not supported by model tests.)

This system does not admit air under any circumstances. The intake water is free from entrained bubbles. Because of the hydrostatic pressure at this depth, the pump could work at higher thrust without serious cavitation than could the installation at the surface (paragraph 5.8.1).

Larger jets, of course, could attain better efficiency. Accommodation of the two 18-inch units inside the vehicle has been at a sacrifice in accessibility of drive components, and an increase in diameter would aggravate this difficulty. Larger jets could be installed in the bottom if the vehicle were driven by a front sprocket. The front drive, although feasible, is not the

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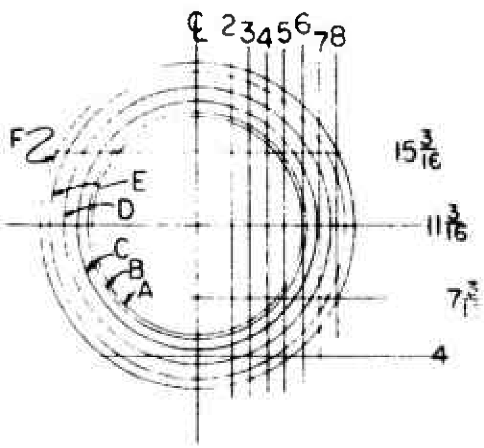

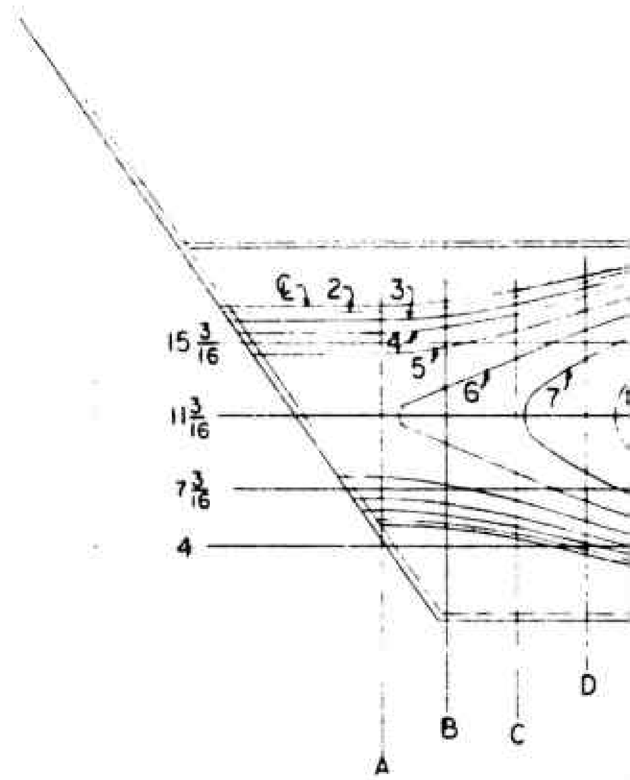

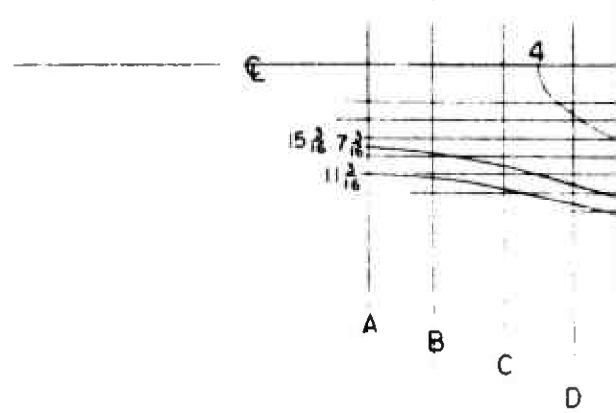

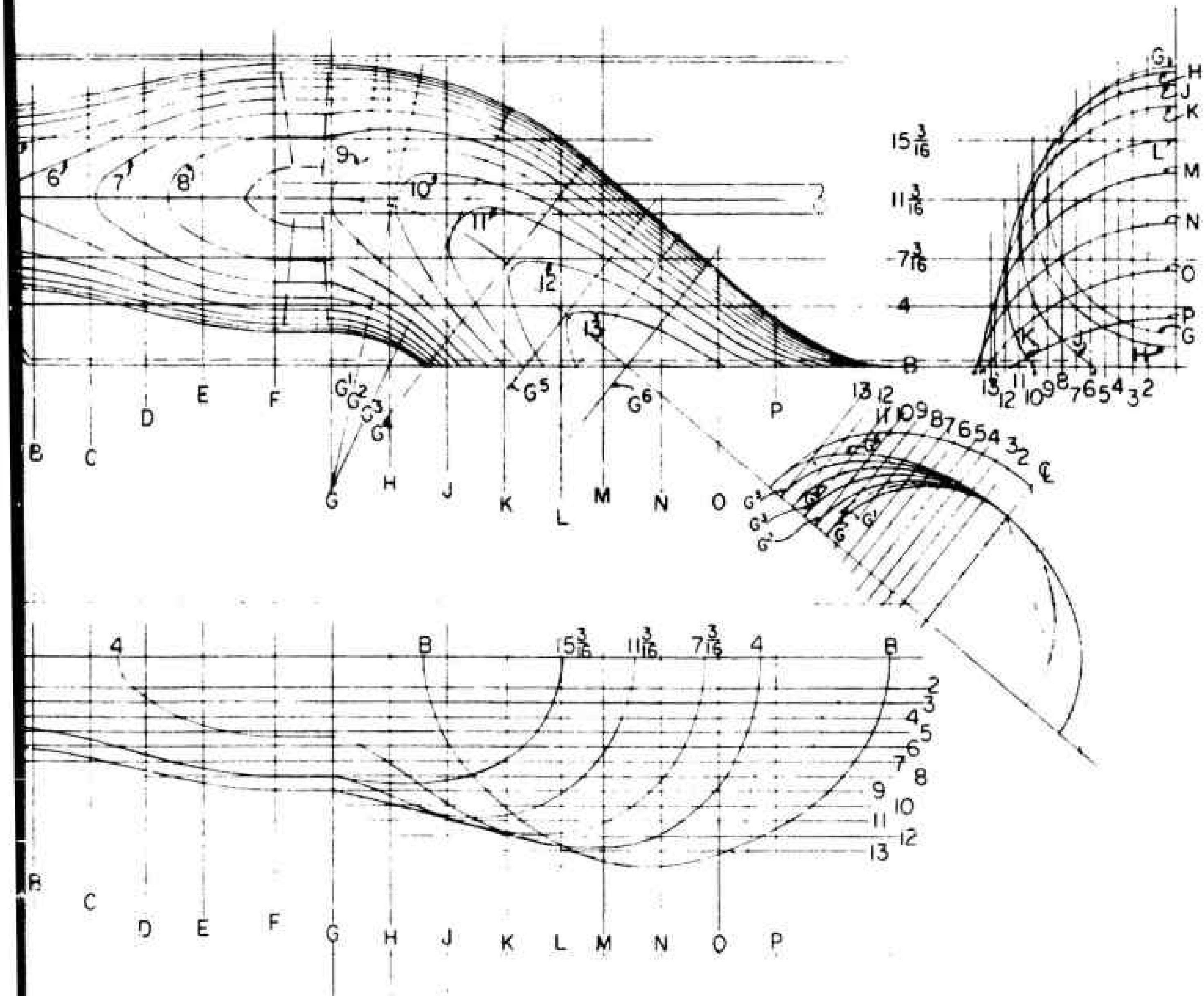

FIGURE 5-8 JET DUCTS FOR A BOTTOM INSTALLATION

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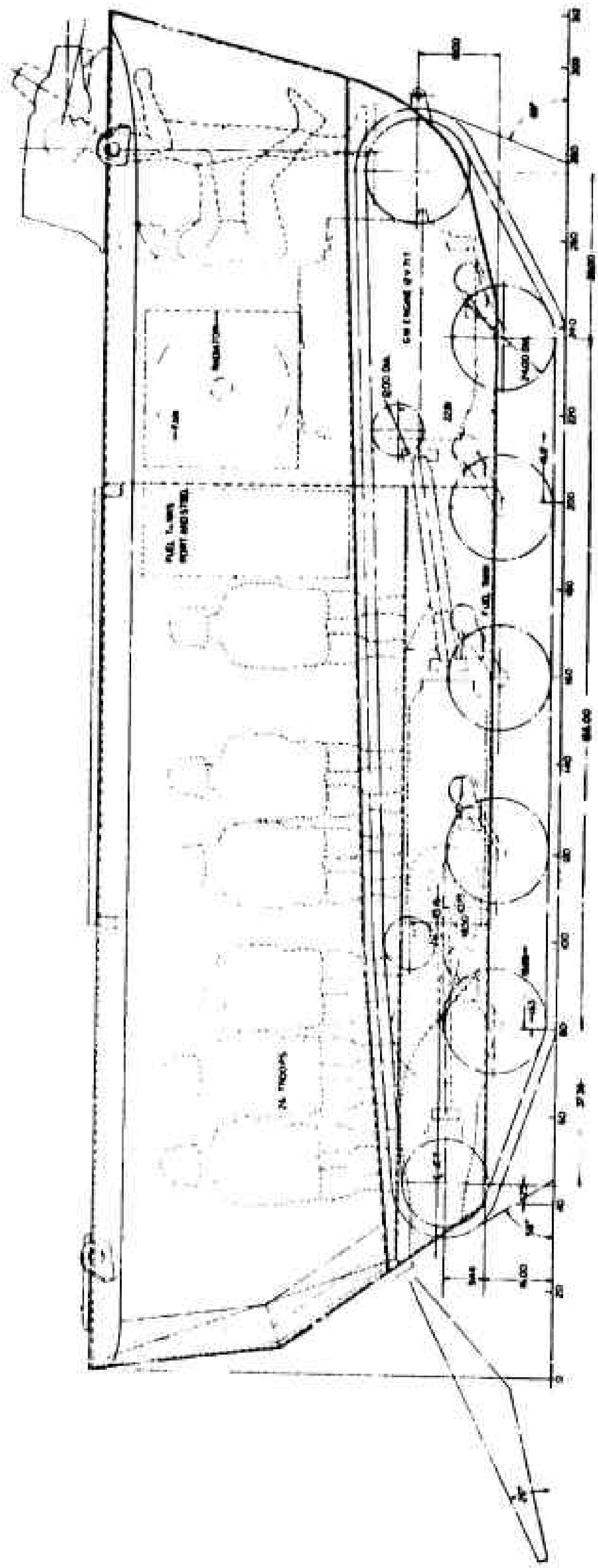


Figure 5-9 LVTPX12 With Bottom Jets - Elevation View

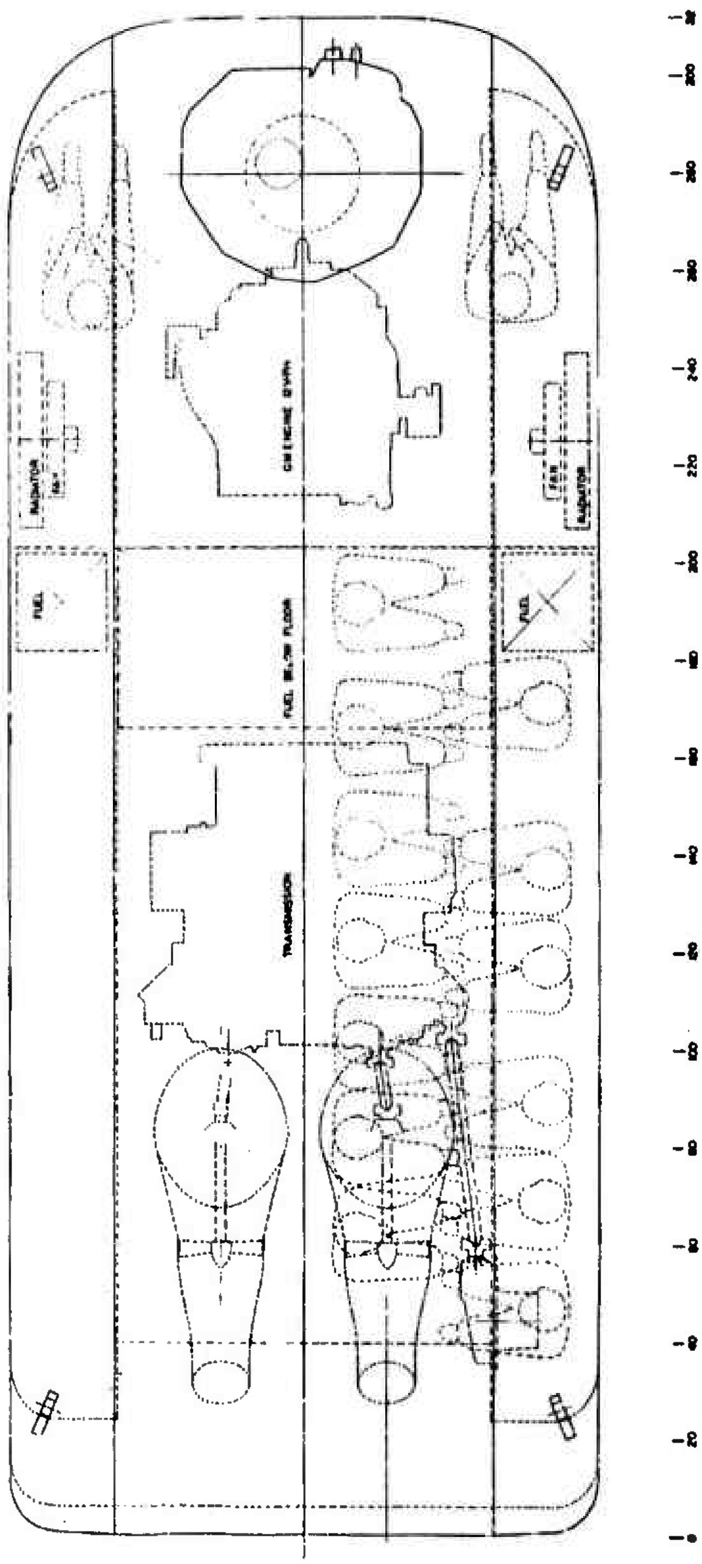


Figure 5-10 LVTPX12 With Bottom Jets - Plan View

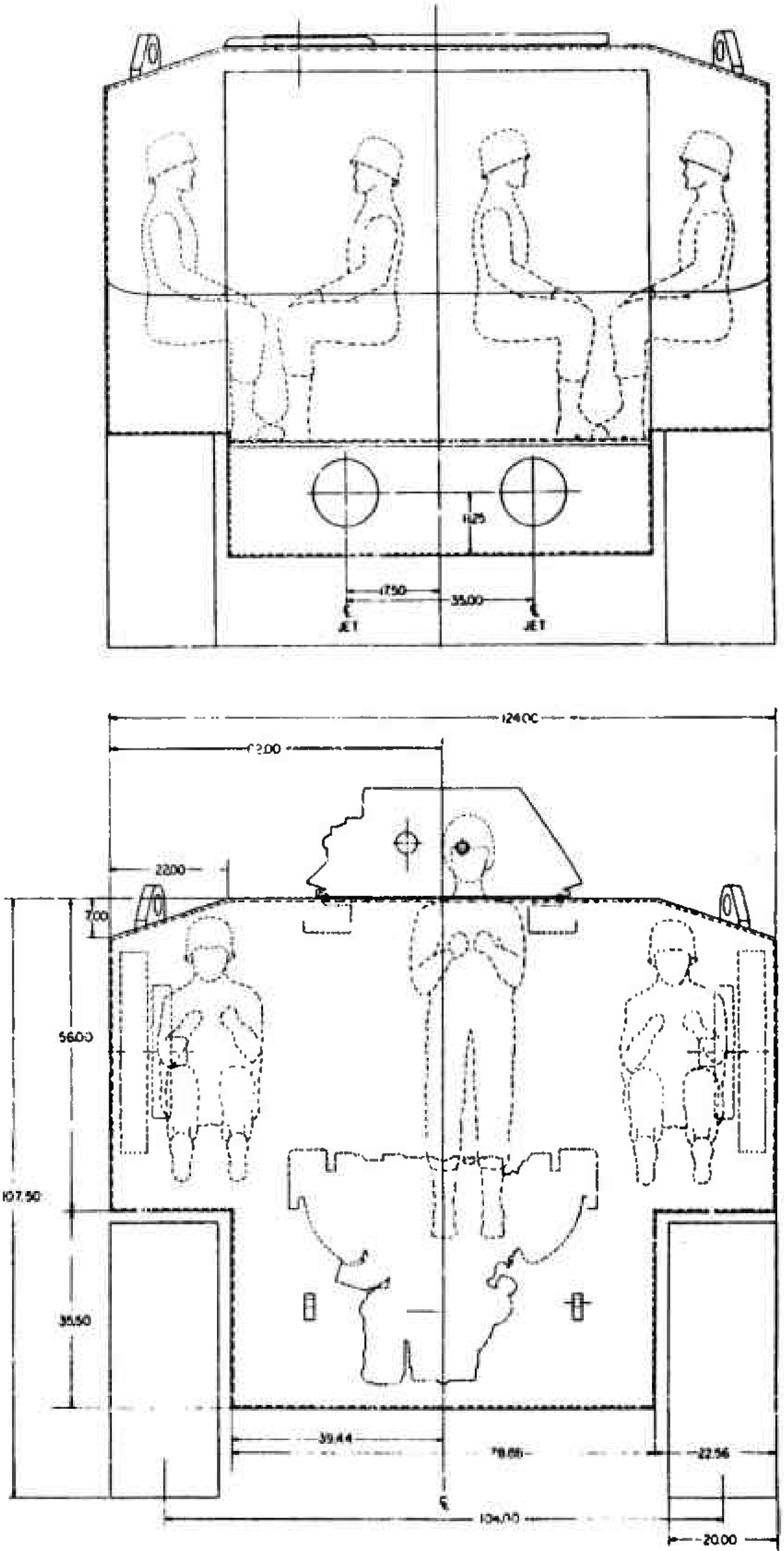


Figure 5-11 LVTPX12 With Bottom Jets - End Views

optimum for meeting obstacles on land. Jets large enough to match the efficiency of the laterally retractable propellers are not feasible in the LVTPX12.

The vehicle of course can be steered and maneuvered by hydrojets, and for this purpose it has been recommended that in Phase II a pair of very small hydrojets be installed on a prototype to investigate the benefit of their use for reactive steering on the track propelled LVTPX12, but it is impossible for jets to match the stopping, backing, or turning power of controllable pitch propellers.

5.8.3 A Hydrojet System With Bow Intake. No designs for any systems of ducts leading from the bow of the craft have been made. If it appears that a jet system taking water from beneath the bow wave ought to be profitable, the following facts should be observed:

- The hydraulic losses from friction in the ducts would be large.
- The size of the ducts would have to be so great that little room would remain in the vehicle for anything else.
- To suck away the bow wave would require a pump moving approximately 100,000 gallons of water per minute, if the craft is travelling at 10 MPH.

Without some probable benefits, there is no point in going through the exercise of arrangement of such a system.

5.8.4 Conclusions on Arrangement With Hydrojets. Neither side jets nor bottom jets add any obstacles to providing the required spaces in the LVTPX12. They do not interfere with providing the stern ramp or the boat bow.

Although the side jet takes up a long distance above the sponson, the necessary cargo space can still be provided, and the specified number of passengers can still be seated. The bottom jets would require some raising of the floor, but adequate headroom would still remain. In fact, somewhat larger impellers could be fitted into the bottom if it were not for the clearance necessary for the final drive and if required intake area could be supplied. Adequate intake area is the first principle of hydraulic efficiency in any pump design.

With either installation, the virtual weight of the vehicle would include the weight of water within the system. This, however, would not be much greater than the lost buoyancy in propeller wells.

Jet propulsion does not interfere with the attainment of the desired features in the LVTPX12 so far as arrangements are concerned. The only reason for rejecting this system is low propulsive efficiency and the unreliability of predictions from tests on models, as analyzed in Appendix A.

5.9 Arrangement With Cycloidal Propellers. A description of the principles of cycloidal propellers is given in Appendix A together with the problems in arrangement of such a system. The chief difficulty is in the design of a system for retracting the blades, whether the propellers act vertically or horizontally. If they are in the bottom of the vehicle, they must be brought inboard for land operation, or even when approaching a reef from the sea. Acting horizontally, attached to driving sprockets, they would not be vulnerable during landing, but would have to be retracted by some means for land operation. The retraction system would be exceedingly complicated. Any possible design would be inconvenient in operation and of doubtful reliability. No detailed designs of any cycloidal system have been attempted (see Appendix N).

5.10 Electric Drive. The various types of electric drive are discussed in Section 6.0 of Appendix D and the conclusion is that the AC system with stator motor control is the lightest, simplest, and smallest. Electric drive offers the ultimate in power plant and power train flexibility since there are no rigid connections between the engine (or engines) and drive sprockets.

The demonstrated success of electric drive in heavy duty construction machinery makes it obligatory that this means of propulsion be given thorough examination for the LVTPX12. The analysis here is not on the engineering of such a system --- that is covered in Section 8.0 and Appendix D --- but on the choice of arrangements that would result if electric drive were adopted.

5.10.1 Effect of Electric Drive on Geometry. The hull shape is not determined by the power train when electric drive is used. The engine and alternator can be wherever convenient as can the drive motors. The engine and alternator can be in the bow and the motors can be in the stern under the floor or vice versa. The interior space and geometry are affected by the location of the components. The alternator is about 1/2 the size and weight of the engine driving it, and as a consequence markedly affects space and trim. The motor diameters determine the floor height required. The more powerful the motors the greater the diameter and the higher the floor has to be.

5.10.2 Electric Drive and Power Train Arrangement. Besides flexibility of arrangement, electric drive offers a simpler power train - no mechanical links between engine and drive sprocket required; simpler cooling system - most of the heat is rejected in external resistors located on the hull; smoothness of drive; simplified controls; and very little sensitivity to temperature, either

hot or cold. An important disadvantage for the LVTPX12 is the lower efficiency of electric drive, 77 percent overall compared to mechanical systems that are over 90 percent. Figure 5-12 shows the delivered horsepowers for electric drive compared to hydrostatic and mechanical transmissions. The electric drive efficiency may be competitive with mechanical when outboard propellers are considered because the motors can be in the propeller hubs, eliminating the complex drive train under certain circumstances. See paragraph 5.10.3. A floating vehicle is very sensitive to weight and the electric drive is much heavier than the other transmissions. Figure 5-12 illustrates this.

The sprocket drive motors can be put under the floor of the track propelled vehicle only if a higher floor or multiple motors are allowed. Normally only two motors are used, one for each track, and the speed difference between the two provides the steering function. For the track propelled vehicle single motors per sprock would approach 20 inches in diameter. The diameter can be decreased by increasing the number of motors. Two motors for each side would have diameters of 16 inches but a combining gear train would be required. In addition, the two motors would weigh mora than the one motor they replace.

The combination of engine and alternator weighs 50 percent more than the engine alone and the length would be increased more than 50 percent. Engine length is critical in the rear ramp configuration and the alternator-engine combination makes it even more critical. For a low powered vehicle the engine and alternator could be installed crosswise in the vehicle. The alternator could be put beside the engine and connected to the engine with a chain or gear drive. This increases the width, the weight, reduces efficiency slightly, increases the complexity, and increases the cost.

PROPULSION MEANS	ENGINE	TRANS.	GHP	DHP	WATER SPEED (MPH)	TRANS. & FINAL DRIVES DRY WEIGHT (LB)
TRACK	12V71T	MECH. (CCS)	800	637	8.1	1500
TRACK	12V71T	HYDROSTATIC	800	563	7.8	2065
TRACK	12V71T	ELECTRIC	800	511	7.5	3130
TRACK	8V71T	MECH. (CCS)	530	394	7.2	1500
TRACK	8V71T	HYDROSTATIC	530	348	6.9	2065
TRACK	8V71T	ELECTRIC	530	313	6.6	3130
PROPELLER	12V71T	MECH. (CCS)	800	680	10.7	1500
PROPELLER	12V71T	HYDROSTATIC	800	676	10.7	2065
PROPELLER	12V71T	ELECTRIC	800	583	10.4	3130
PROPELLER	8V71T	MECH. (CCS)	530	443	9.8	1500
PROPELLER	8V71T	HYDROSTATIC	530	439	9.7	2065
PROPELLER	8V71T	ELECTRIC	530	381	9.4	3130

Figure 5-12 Drive System Comparison

Electric drive permits multiple engine installations but, if the two alternators feed common motors, the alternators must be synchronized. This is a very complicated process, more complicated than synchronizing two engines for a mechanical drive. The synchronizing problem can be eliminated if each alternator feeds only one motor and there is no electrical tie between the systems. Steering is accomplished the same as before except that the steering is not regenerative. In this case, failure of one engine would fail the entire vehicle. Transmission cooling for electric drive is simplified since the greatest heat rejection is in the external resistors outside the vehicle. The cooling of the components themselves can be accomplished by ventilating the space as is done with a mechanical transmission having air cooled brakes.

Service on electric drive is simplified due to the reduced number of components. Men trained on ordinary electric motors and controls can service the electric drive.

The component prototype costs are less than for a new transmission and the pieces can be available much sooner, within 6 months for electric drive rather than 12 or more months for a new transmission.

The controls are simplified since only one control is needed for each motor that serves the function of direction, speed, and braking. A mechanical brake is incorporated to serve as an emergency device and for parking.

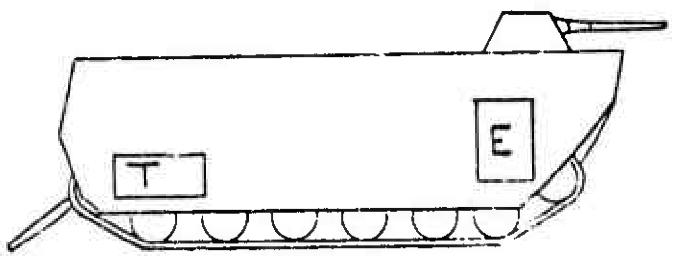
The LVTPX12 with electric drive has the option of eliminating all hydraulic elements and using electricity to power all accessories. Low temperature problems and leaks are eliminated. Each vehicle also can be the power source for an arc welder.

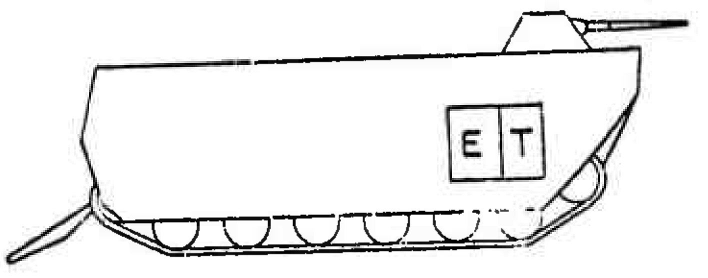
5.10.3 Electric Drive and Auxiliary Propulsion. An electric drive appears convenient for driving auxiliary propulsive devices. If the driving motors are independently variable in speed, the thrust produced by each of twin screws can be independently controlled, thus providing for steering, maneuvering, stopping, and backing. If the driving motors are inside the pod in front of the propellers, bevel gears and shafting to the outboard units are eliminated, and furthermore the water flowing around the motor pods can be used for cooling. But variable speed motors to drive the propellers of the LVTPX12 would have to be 18 inches in diameter. In front of 24-inch screws, these motors would result in a very high thrust deduction and very low propulsive efficiency. On the other hand, high speed motors, which could be only 14 inches in diameter, would have to be constant speed. Therefore, if the smaller high speed motors were placed outboard with the propellers, the controllable pitch feature would be the only way independent control could be obtained. Electric drive is consequently not so rewarding as it appears: either use large, heavy, variable speed motors driving fixed pitch propellers through shafting and bevel gears, or use constant speed motors driving controllable pitch propellers directly.

5.10.4 Electric Drive Conclusions. In summary, the electric drive offers advantages in flexibility of power train component location, prototype availability, program costs, accessory power options, and control simplicity. Its prime disadvantages are component sizes, weight, and efficiency. The latter three reasons are overriding at present for the LVTPX12 and therefore electric drive is dropped from further consideration.

Arrangement With Single Engine. None of the concepts studied so far have been found to be without serious objections in weight, efficiency, water speed, or effect on available space. While most of these objections cannot be applied to a single engine installation, it has a few acknowledged deficiencies of its own. It will be shown, however, to be the best bargain in the long run when cost, reliability, efficiency, and simplicity are weighed together. The merits of a large number of suitable engines are compared in Section 8.0. The GM 12V71T is selected as the lightest of available engines above 530 horsepower.

5.11.1 Location of a Single Engine. The boat bow and the stern ramp, already agreed to be highly desirable, require the single engine to be forward. The general arrangement looks as follows:





With the second of the above arrangements, weights are concentrated forward. Since this is obviously undesirable, the transmission must be in the stern. The further consequence is a rear sprocket drive. (The merits of front and rear drives are examined in Section 9.0.)

The transmission to match the 800 horsepower engine is shown in Section 8.0 to be the Chrysler CCS-1, flat enough to fit under the cargo floor of the LVTPX12.

5.11.2 General Effects of a Forward Engine. A mixture of miscellaneous effects results from adopting the stern ramp with engine forward. Most of the objectionable ones are engineering challenges rather than incurable defects, and there are some definite advantages.

- Silhouette as determined by floor height: High. The height is relative, of course, caused by the necessary depth for the transmission. The inside headroom, however, is not the final criterion for height, but rather, the submergence of the bow at high speeds. It is true that if water speed is not selected as the prime aim, the deck can be lowered; but, in this case, water speed has been so selected.
- Vertical center of gravity: Low. The low engine, transmission, and fuel insure high stability both on water and on land.
- Freedom in bow design: Maximum. Any bow suitable for land operation can be adopted.
- Space for driver: Ample. He is not restricted, either fore and aft or athwart ships.

- Space for gunner: Maximum. He is slightly ahead of the engine. He can stand up. Any convenient fixtures can be installed for his efficiency. He does not occupy useable cargo space and does not interfere with embarked troops.
- Accessibility of machinery: Mostly good, partly bad. The engine is accessible at all times. The transmission may be covered by cargo.
- Relative weight: Neutral. The only extra item with engine forward, transmission aft, is a long drive shaft. This is not a large weight.
- Space utilization: Neutral. The drive shaft takes up space that might otherwise be used for fuel tanks. The space over the engine is almost all usable. No space is wasted by a raked ramp.
- Location of cooling system: Side. Satisfactory; no serious amount of righting moment is lost on the radiator side.
- Watertight compartmentation: Very difficult. A watertight bulkhead could be placed between engine and cargo compartments, but would require seals and watertight doors, would interfere with space and communication, and would make engine removal difficult.
- Controls: Mixed. Connection to throttle would be short, but to transmission, long.
- Structural problems: Require solutions. For removal of engine, the gun cupola could be built into a hatch, or else, a plan must be provided for moving the engine horizontally aft and then through the cargo hatch. No other configuration is free from similar problems.

- Drive to propellers (if screw propelled): Mildly inconvenient. A drive shaft must go aft over the track sponsons.
- Interior communication: All personnel, both crew and troops, can see each other. Nobody is isolated.
- Ventilation in passenger space: An engineering problem. The engine cannot exhaust out the rear, but the exhaust can be directed downwards, alongside, eliminating fumes in the neighborhood of the air intakes.
- Engine noise: Unfavorable. While the engine would be enclosed and would not emit any more noise than if it were in its separate compartment, it is near the crew.
- Multisource engines: Not very flexible if weight is to be kept low, power high, and water speed up. If weight or speed are to be conceded, of course, flexibility becomes high. Air-cooled engines, however, will be difficult to use (See Section 8.0).
- Weight concentration and trim. Bow down when light for the screw-driven version, and bow down all the time for the track-propelled version. From the standpoint of water speed, these are not objections.
- Fuel location: Flexible. Tanks may be placed entirely under the floor, or partly under the floor and partly over the sponsons.
- Components exposed to bilge. Engine, transmission, and fuel tanks will be subject to wetting by bilge water.

5.11.3 <u>Conclusions on Forward Engine for Maximum Water Speed</u>. In any case for maximum water speed, a stern ramp with the consequent freedom to design an optimum bow is a practical necessity. The location of the engine thus

is forward, and the transmission is advisedly aft for weight distribution. The arrangement of interior space is no problem, unless the beam is reduced beyond limits that are impractical for other reasons. Although, with this arrangement, there is less freedom to lower the deck, the high speed demands that the freeboard at the bow not be reduced.

5.12 Selection of Concepts. Maximum water speed is undoubtedly a prime choice among the several alternatives that might characterize the LVTPX12. The order of priorities is accepted as the following:

- The specified water speeds.
- The specified degree of ballistic protection.
- The specified maximum exterior dimensions.
- The specified weight.

No other arrangement results in better dimensions, armor, or weight than the arrangement with boat bow, stern ramp, and forward engine. None of them satisfied the demand for maximum water speed as well.

Therefore, the broad description of the LVTPX12 must be as follows:

- A finely shaped bow.
- A stern ramp, as vertical as possible.
- A single, forward engine developing 800 horsepower.
- A transmission under the floor, as far aft as possible.
- A gun turret on the centerline as far forward as possible.

5.12.1 Selection of Characteristics for a Track-Propelled LVTPX12.

A vehicle propelled by tracks alone, to make 8 MPH water speed, conforms to this broad description. The optimum details of this vehicle are described in the appropriate sections of this report. For example, details of an optimum track and suspension system are shown in Section 9.0, selection of armor is covered in Section 6.0, and analysis of structure is contained in Section 7.0.

Arrangement of machinery and spaces is shown in Figures 5-13, 5-14, and 5-15. The outside appearance of the track-propelled LVTPX12 may be seen in these drawings, and a perspective view may be inspected in the frontispiece.

This vehicle is powered by a single GM-12V71T engine developing 800 horsepower at 2500 RPM. A complete description of the entire power train, controls, clutches, transmission, and final drive may be found in Section 8.0.

The hydrostatic characteristics of the track-propelled LVTPX12 may be read from Figure 4-83. In particular, these curves provide the waterline length at any given displacement. Figure 4-79 and Figure 4-83 show the righting moments for this vehicle up to 105 degrees of heel.

A concise description of this optimum version of the track-propelled LVTPX12 is contained in the Summary of Characteristics, Paragraph 5.12.3.

Attention was called in Section 4.0 to the proven necessity of certain appendages for maximum efficiency in propulsion by tracks: bow fenders, side skirts, and stern baffles with short contravanes. Operators have become accustomed to side skirts on all existing tracked amphibians, and these

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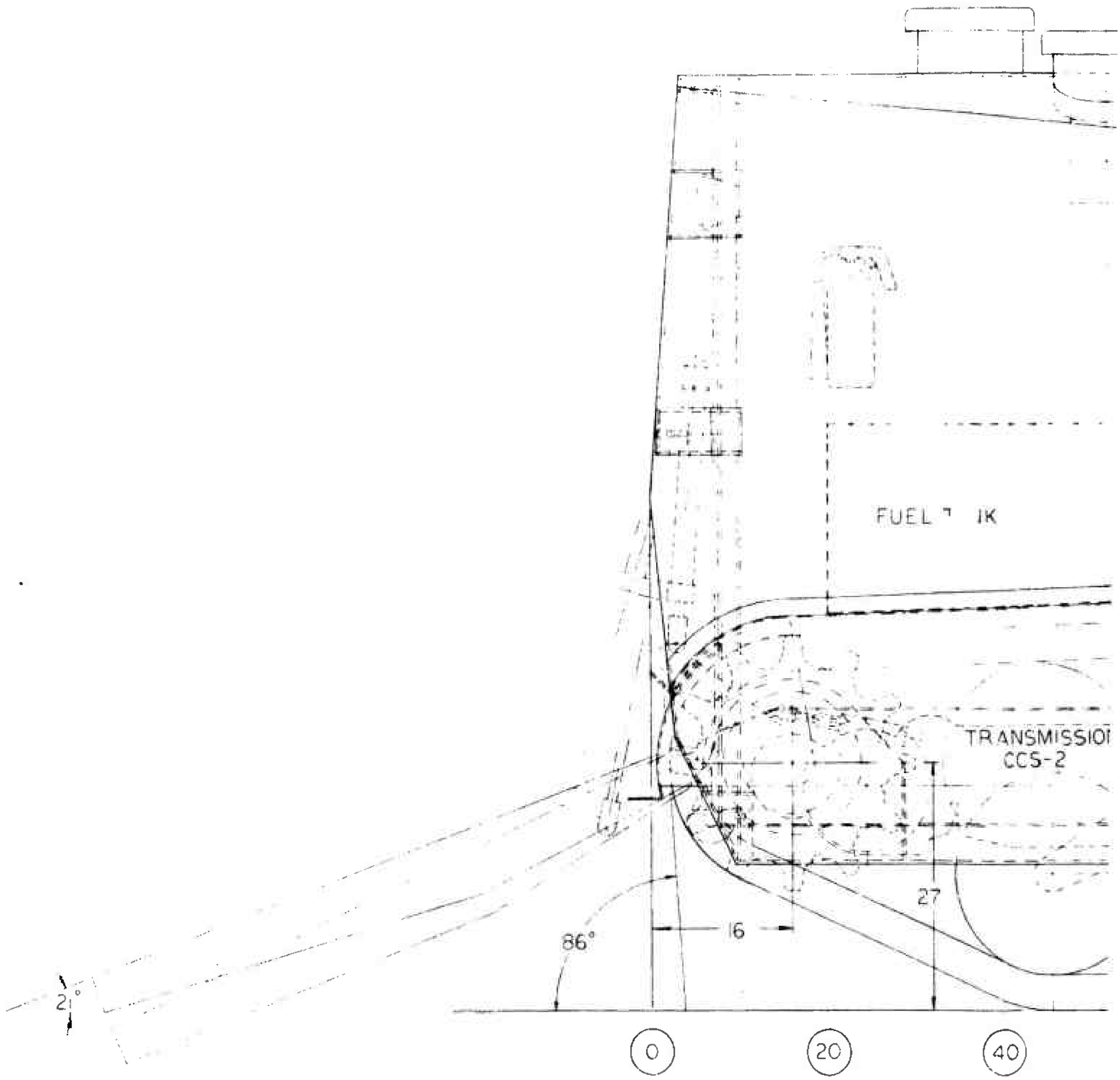
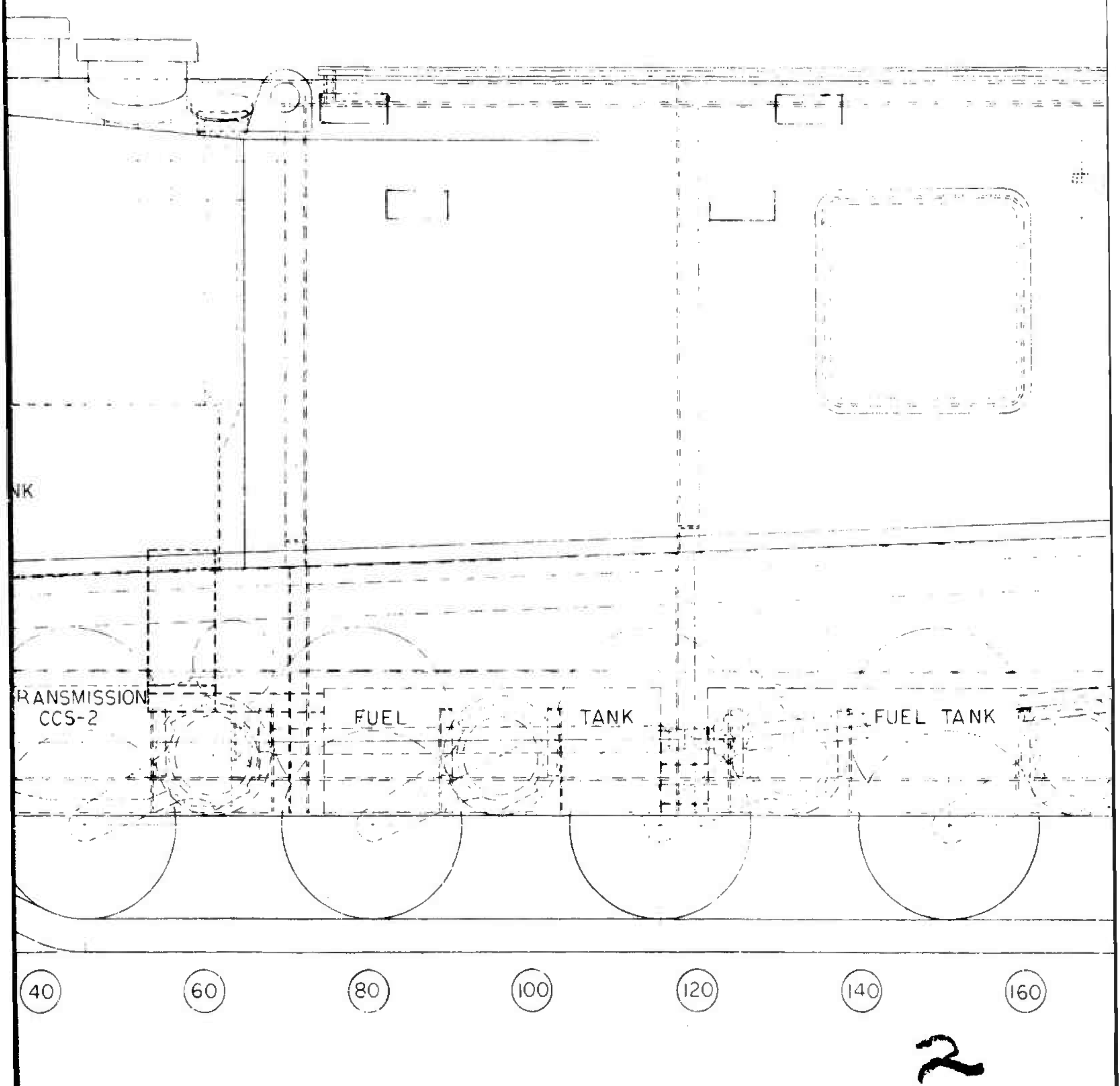
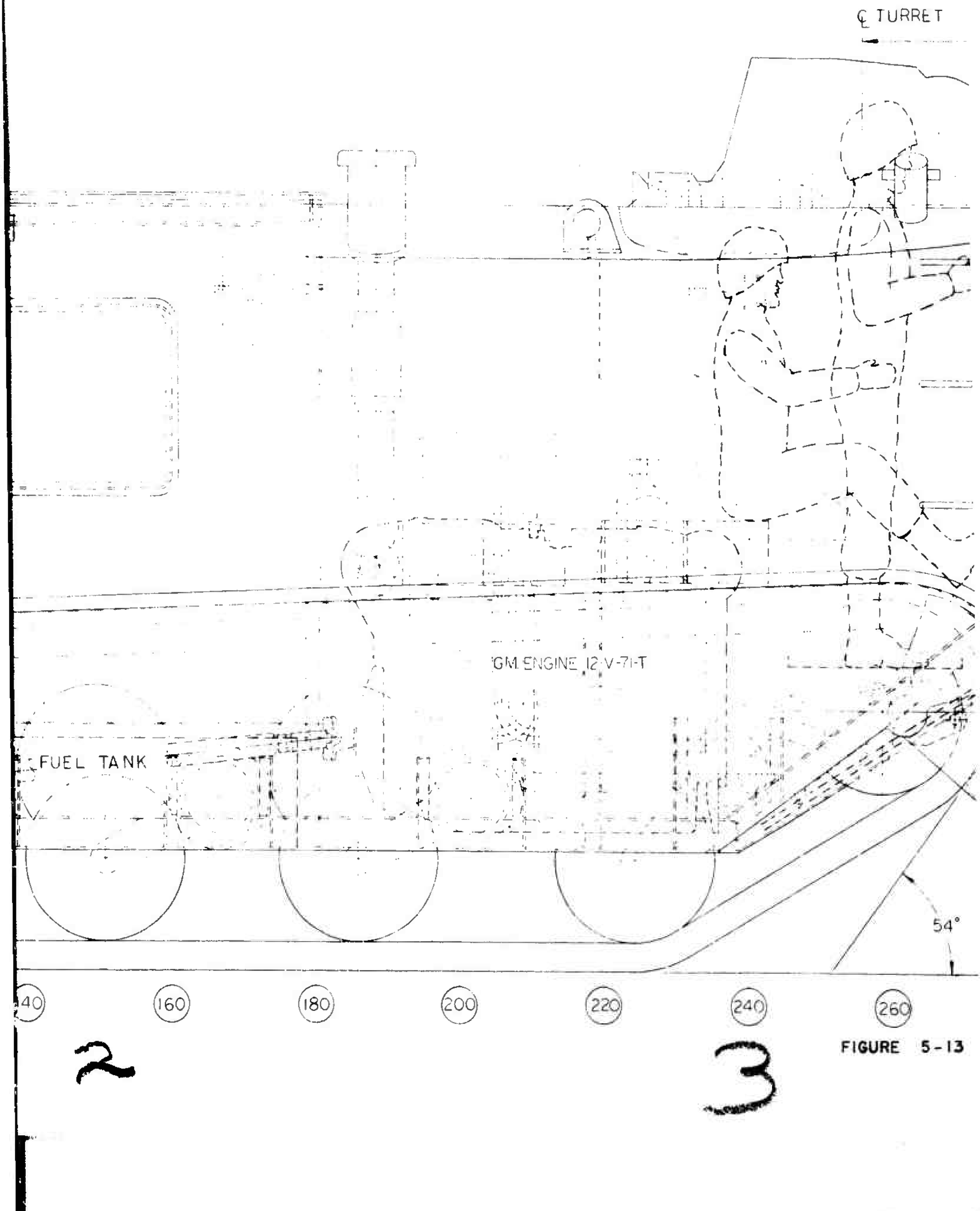




NK
RANSMISSION
CCS-2
FUEL
TANK
FUEL TANK
40
60
80
100
120
140
160
2



℄ TURRET
GM ENGINE 12-V-71-T
FUEL TANK
54°
140
160
180
200
220
240
260
2
3

FIGURE 5-13

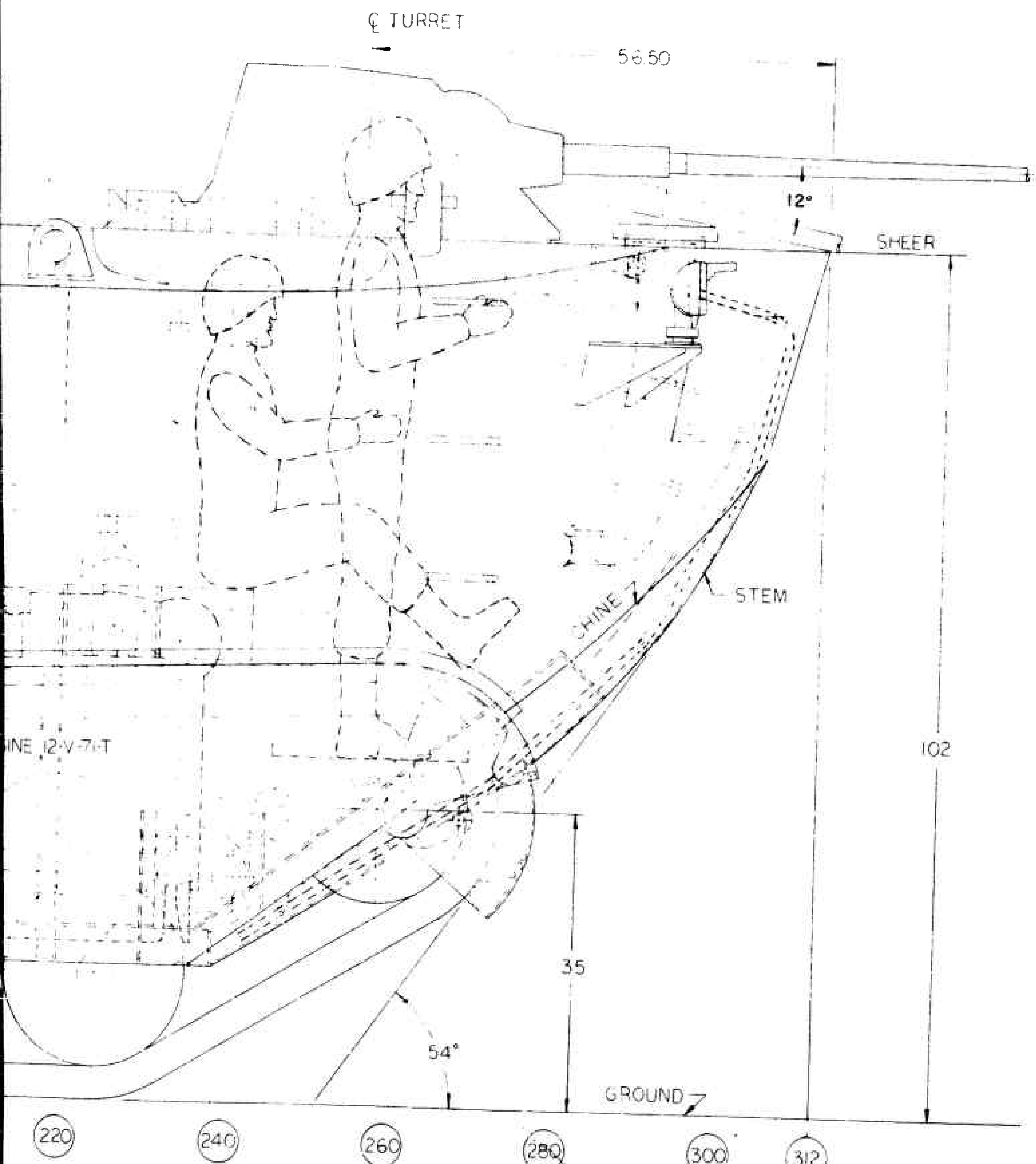


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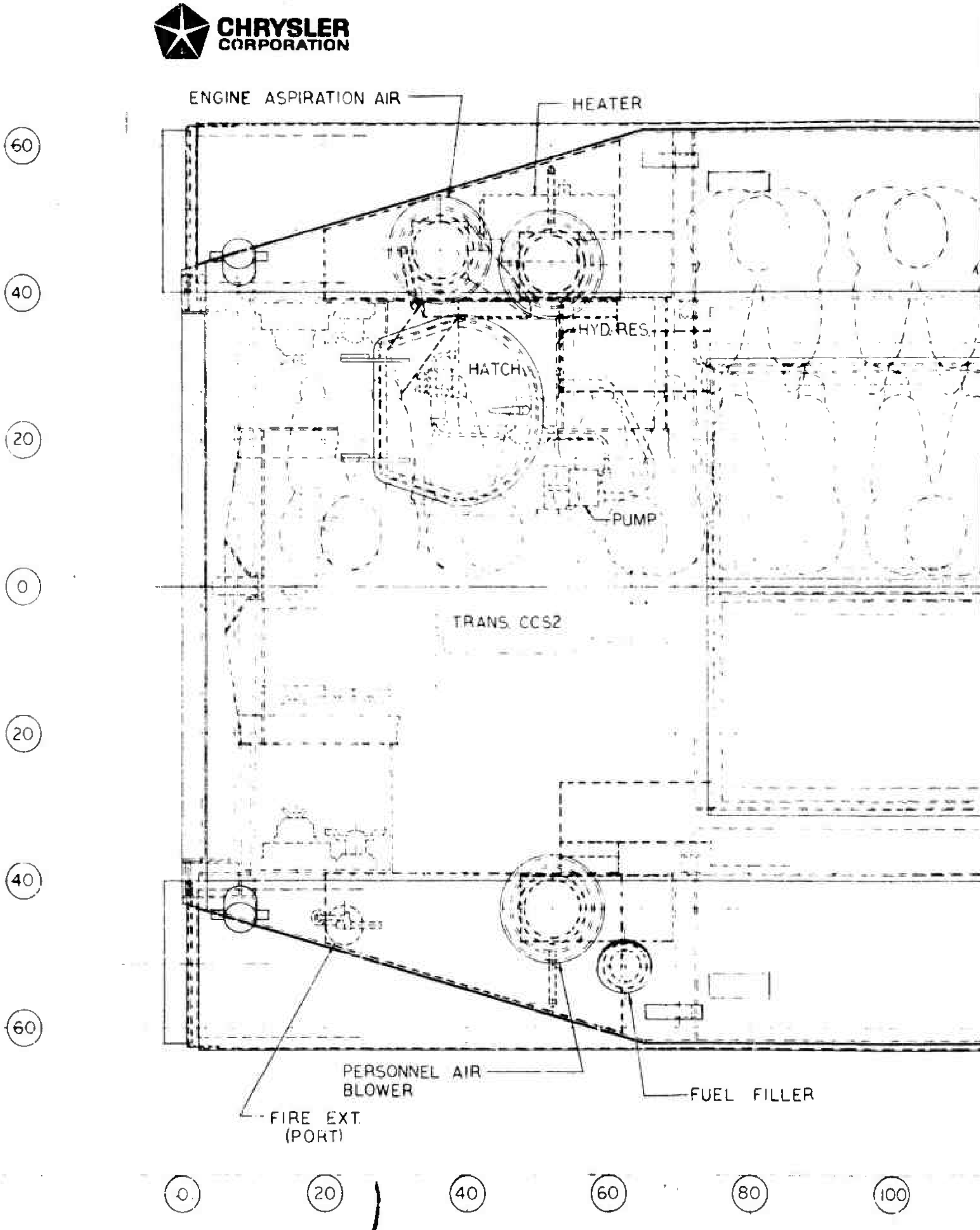
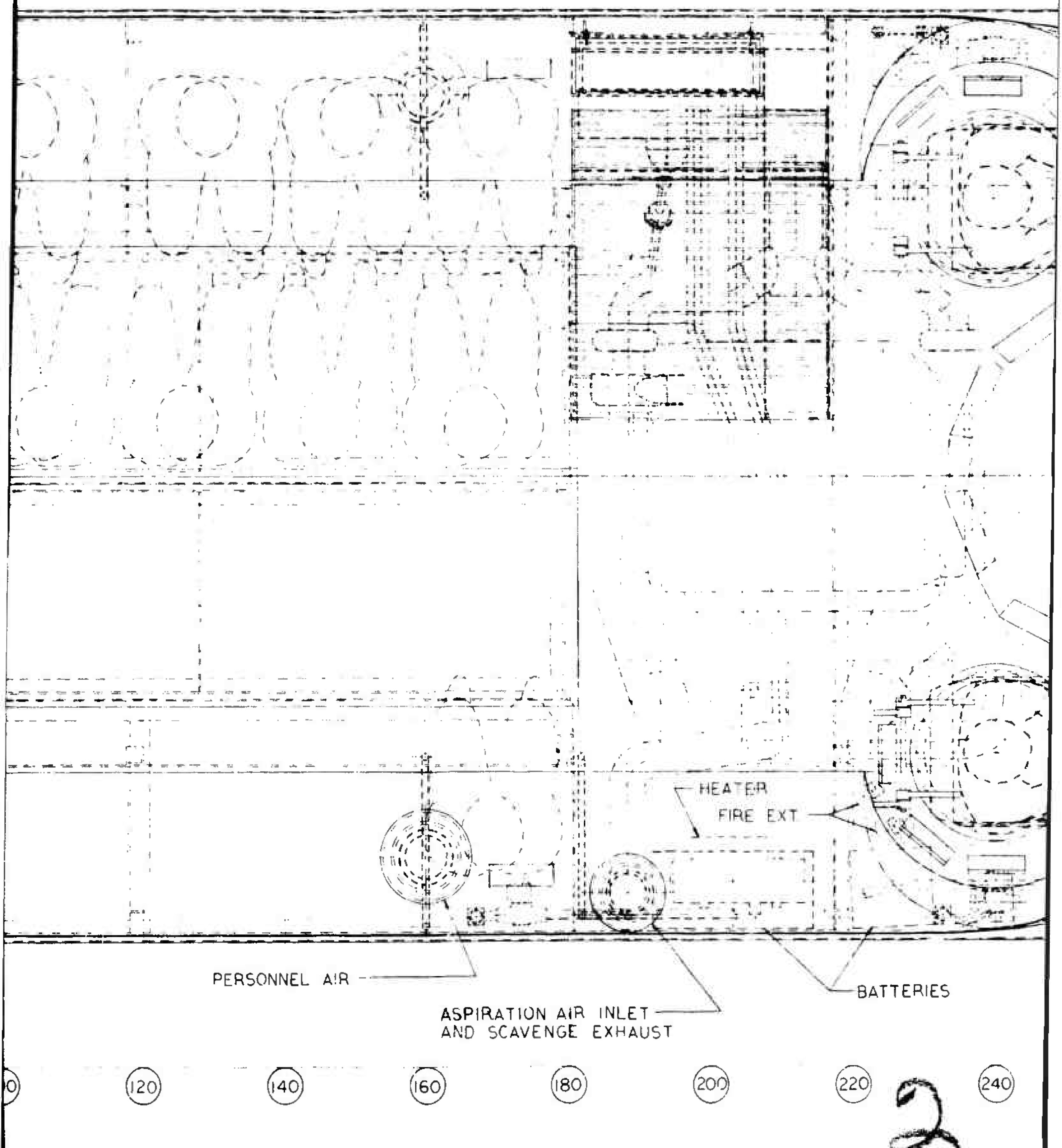
FIGURE 5-13 TRACK PROPELLED LVTPX12--ELEVATION VIEW

4


CHRYSLER
CORPORATION
ENGINE ASPIRATION AIR
HEATER
HYD RES
HATCH
PUMP
TRANS CCS2
PERSONNEL AIR BLOWER
FUEL FILLER
FIRE EXT (PORT)
60
40
20
0
20
40
60
0
20
40
60
80
100


HEATER
FIRE EXT
PERSONNEL AIR
BATTERIES
ASPIRATION AIR INLET
AND SCAVENGE EXHAUST
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160
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200
220
240
2

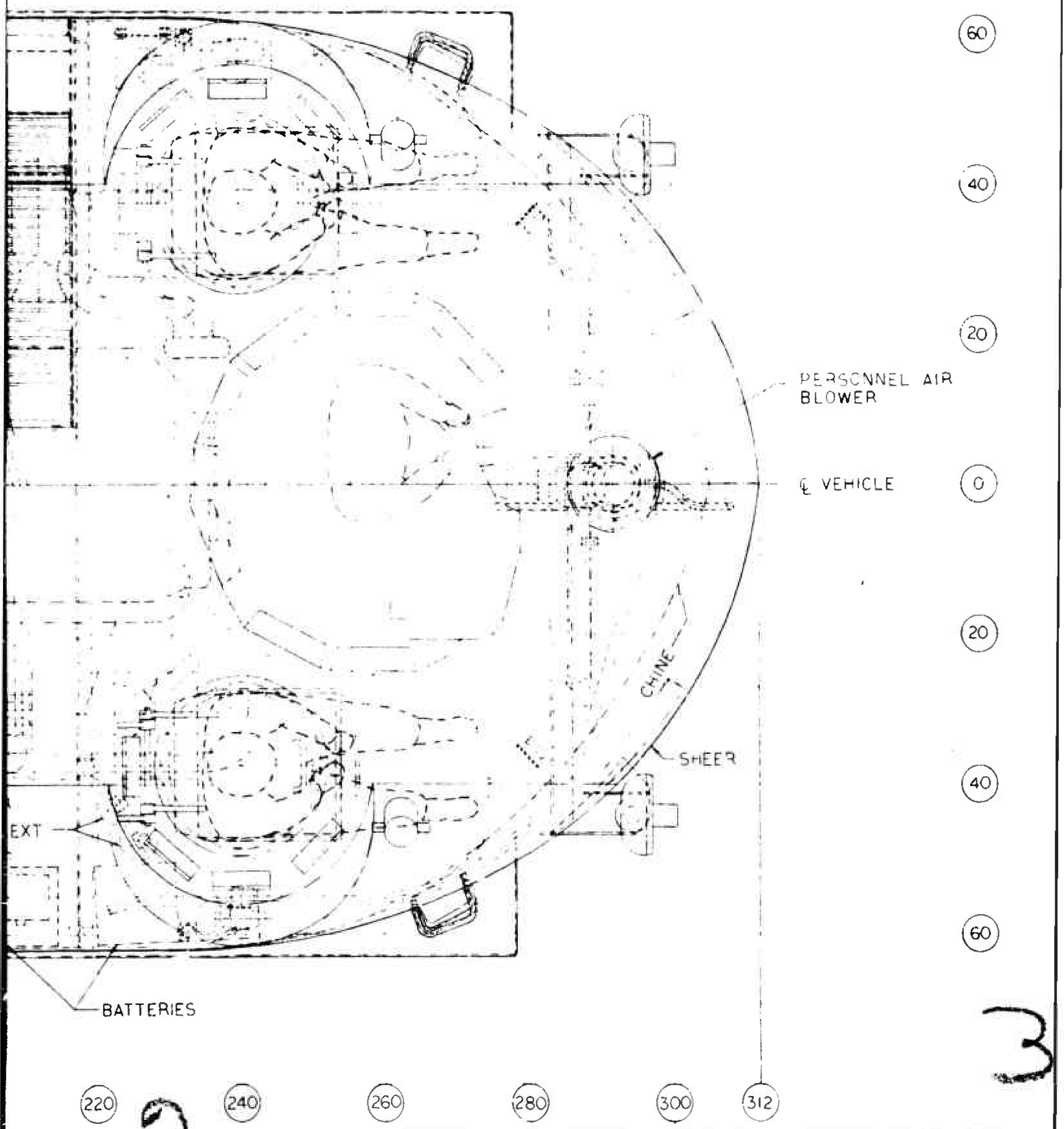

FIGURE 5-14 TRACK PROPELLED LVTPX12-PLAN VIEW

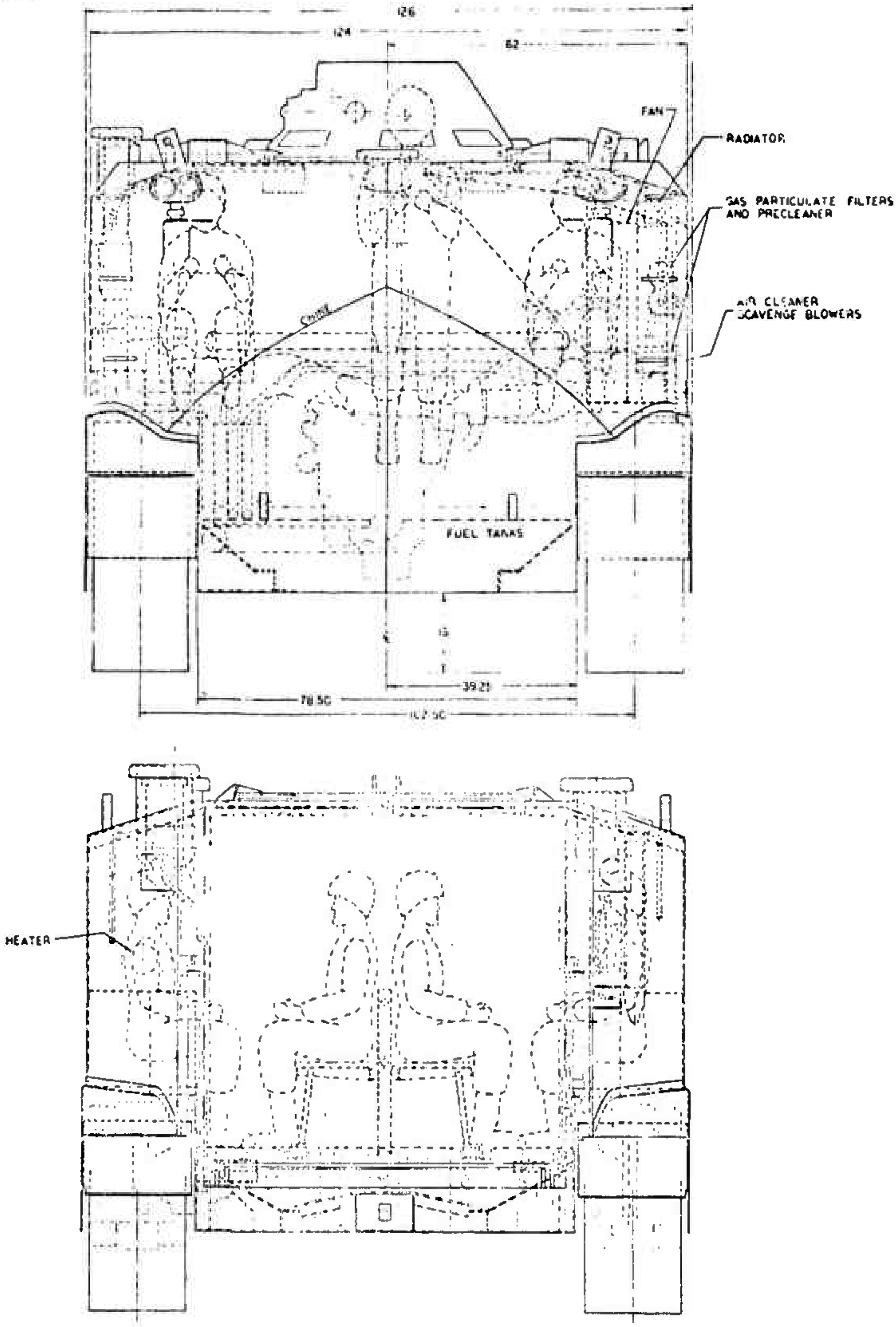

Figure 5-15 Track Propelled Vehicle - End Views

appendages seem to be no serious problems in operation. The bow fender and the stern contravane must be retracted in certain situations on land. The mechanism for retraction of both these devices is shown in Figure 5-12.

A detailed analysis of weights of this design is in Paragraph 5.11, Appendix N. Several selected alternative engines, besides the GM-12V71T, are included in the tables. The power outputs of these engines may be found in Section 8.0, and the consequent performance of the LVTPX12, if equipped with them, may be found in Section 5.2 of Appendix D.

5.12.2 Selection of Characteristics for a Screw-Propelled LVTPX12. The optimum shape of the LVTPX12 for best efficiency in the water is hardly different for screw propulsion than for track propulsion. In fact, the boat bow is exactly the same, the only difference being that wells are cut into the sides at the stern to house the propellers.

The profile, plan, bow view, and stern view are shown in Figure 5-16 through Figure 5-18. The interior arrangement of this version is almost exactly the same as that of the track-propelled vehicle. The only difference is the mechanism for powering the outboard, controllable pitch propellers and the system by which the helmsman maneuvers the LVTPX12 through pitch changes in the propellers. The maximum water speed of this vehicle is in excess of 10 MPH.

On the outside, this screw-propelled LVTPX12 also differs from the track-propelled version in the absence of the bow fenders and the stern contravanes. These are not necessary, of course, for screw propulsion and, in fact, the propellers will do somewhat better without them. Nevertheless, the same

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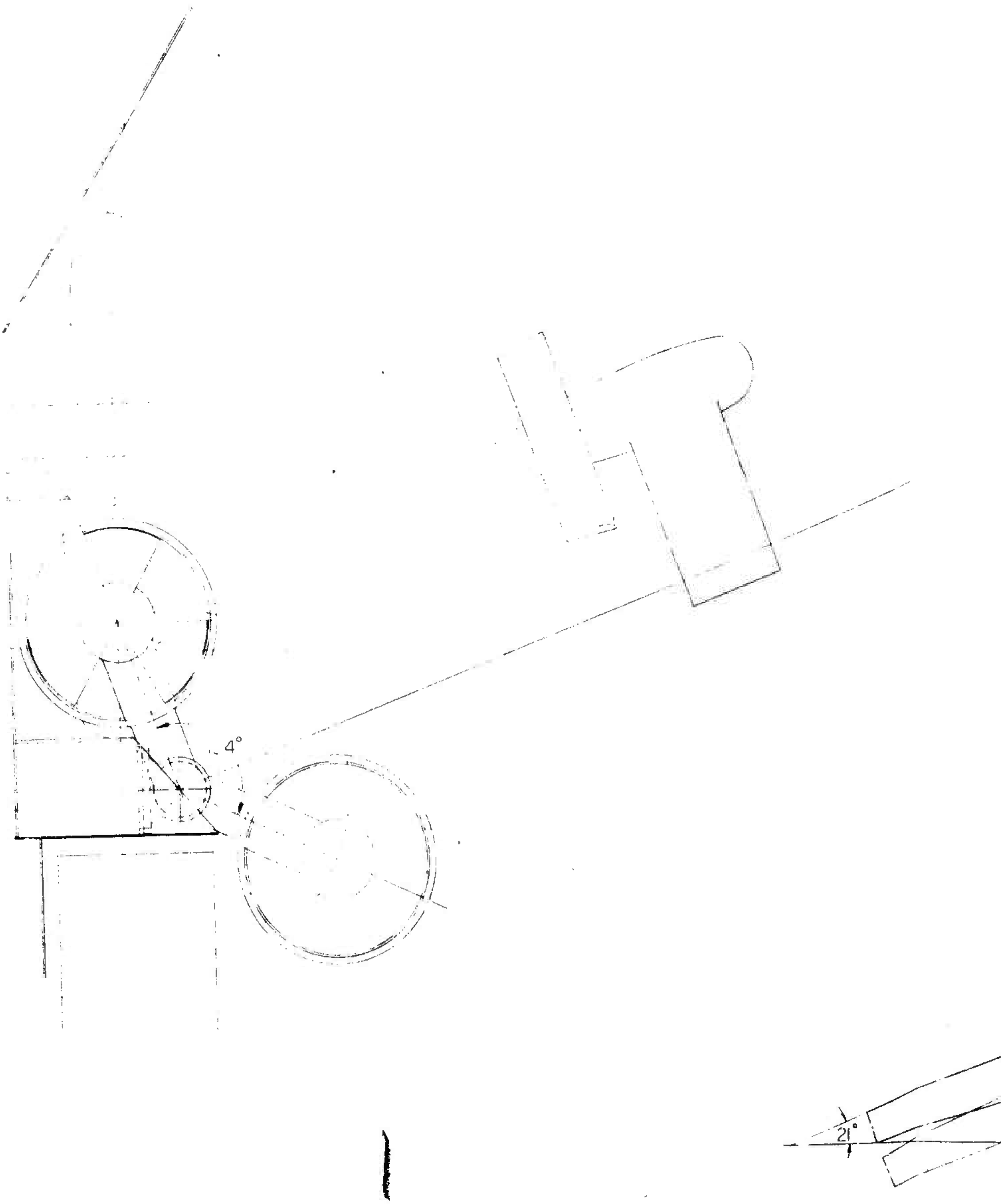
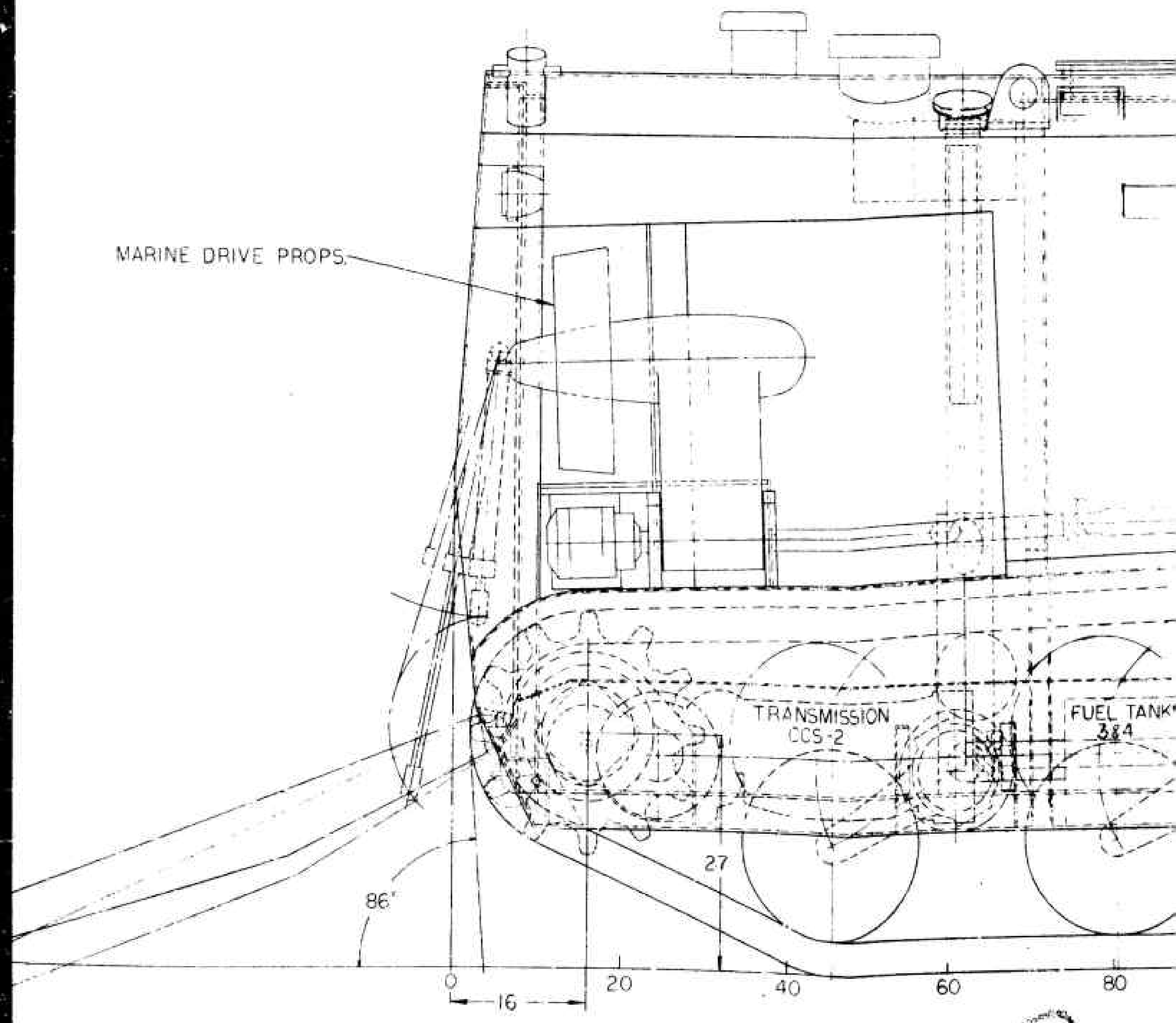
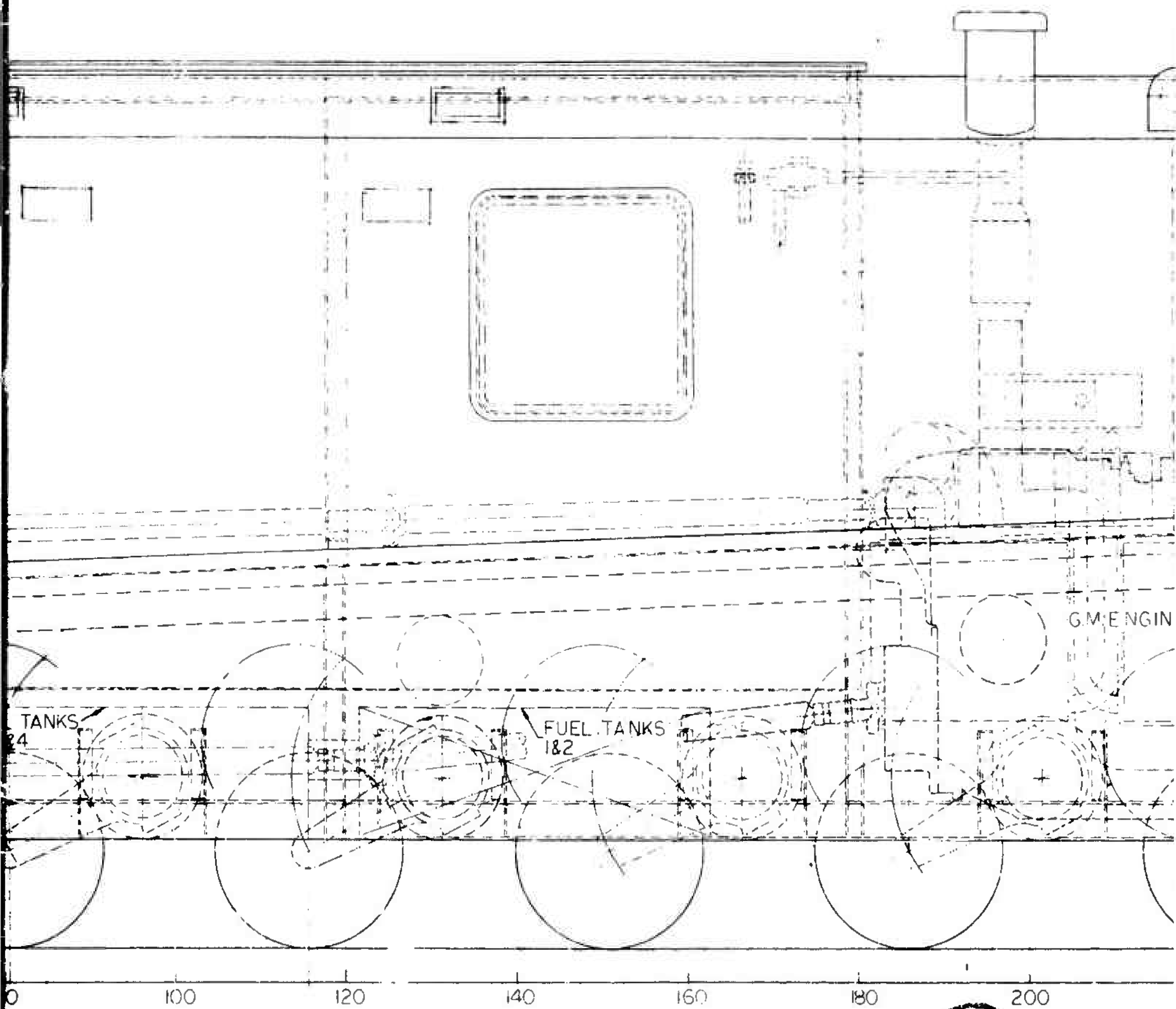



MARINE DRIVE PROPS.
TRANSMISSION
CCS-2
FUEL TANKS
3&4
86°
27
0
16
20
40
60
80



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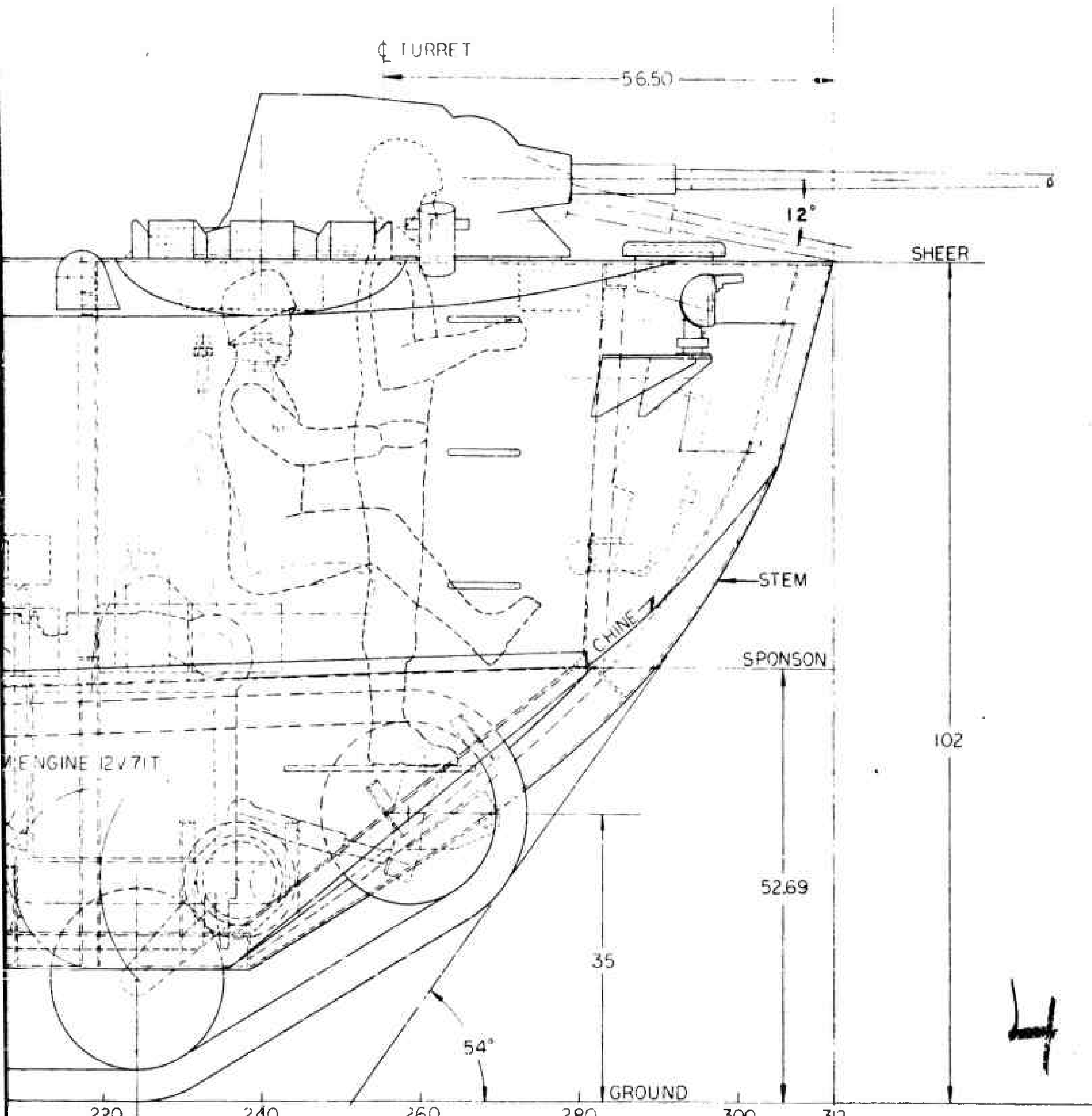

FIGURE 5-16 SCREW PROPELLED LVTPX12-ELEVATION VIEW

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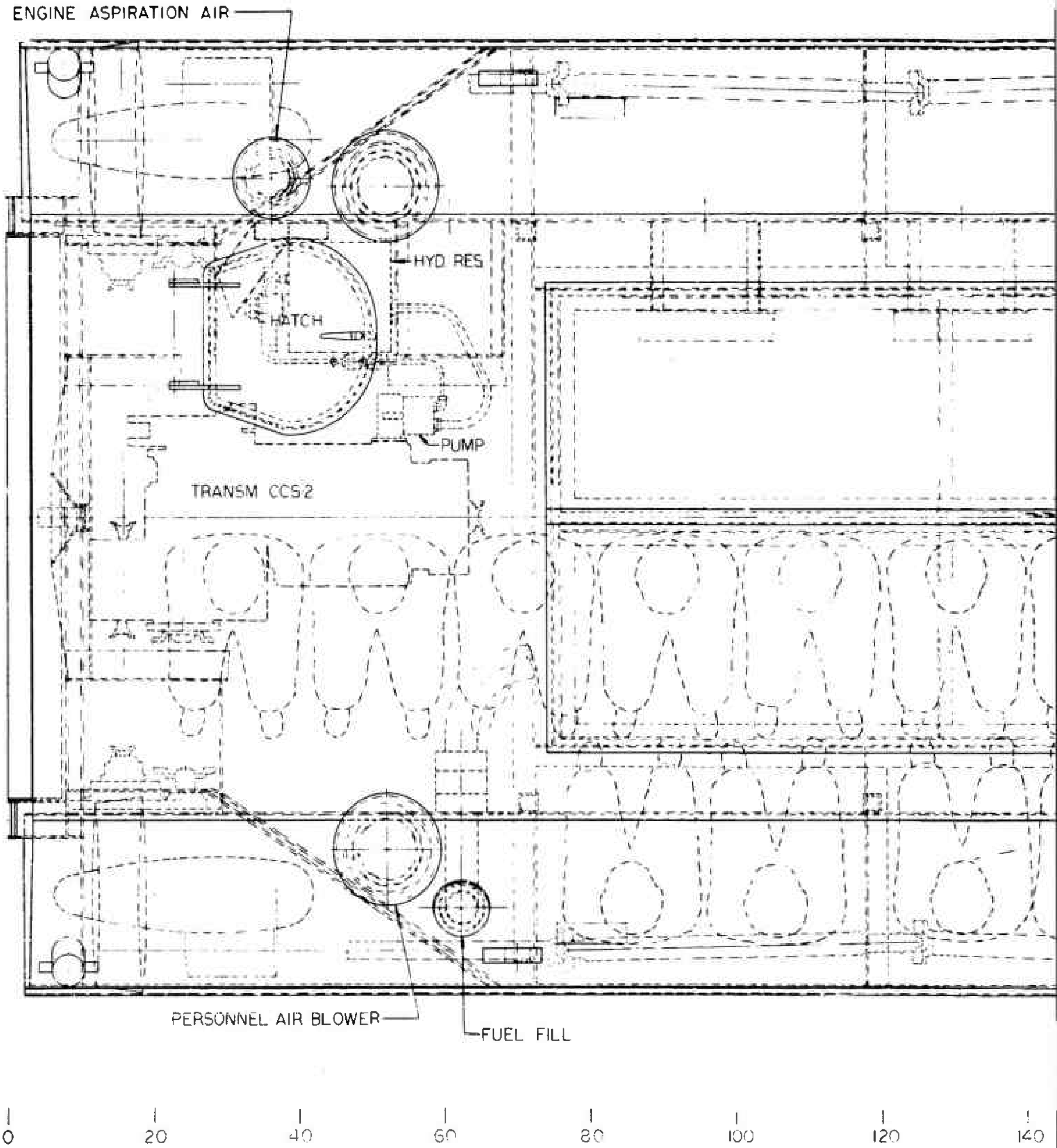


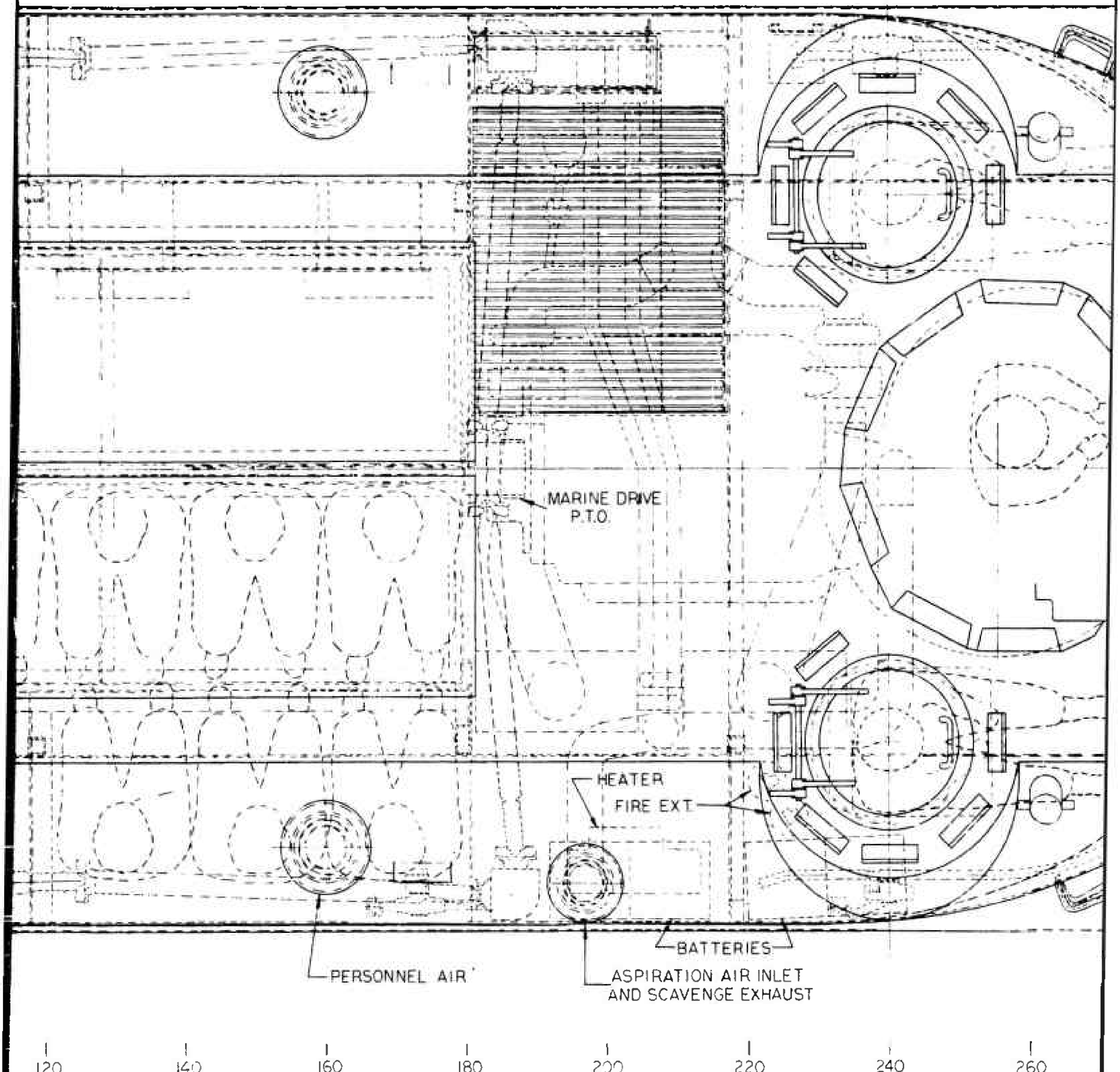


FIGURE 5-17 SCRE

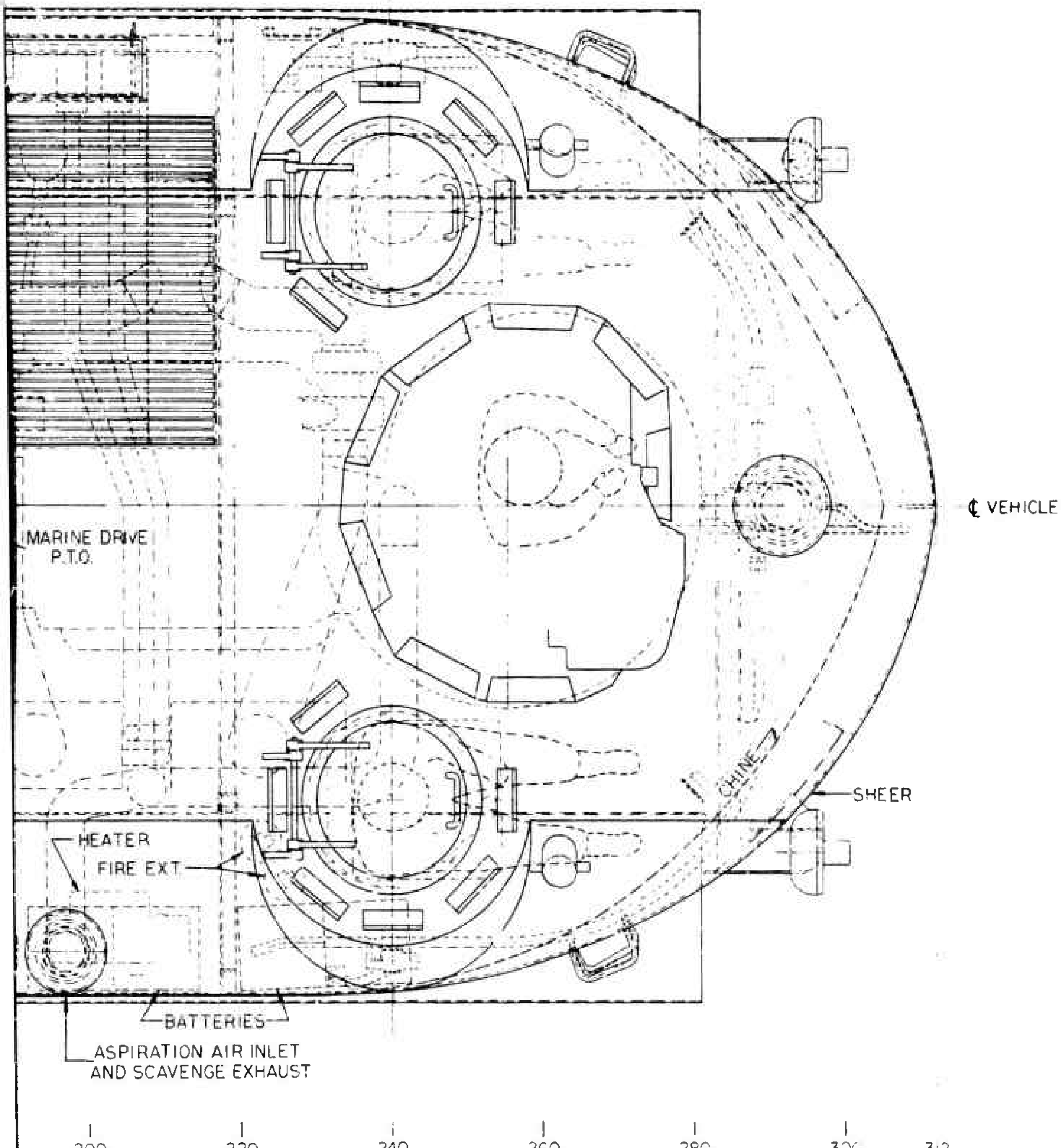


FIGURE 5-17 SCREW PROPELLED LVTPX12 — PLAN VIEW

3

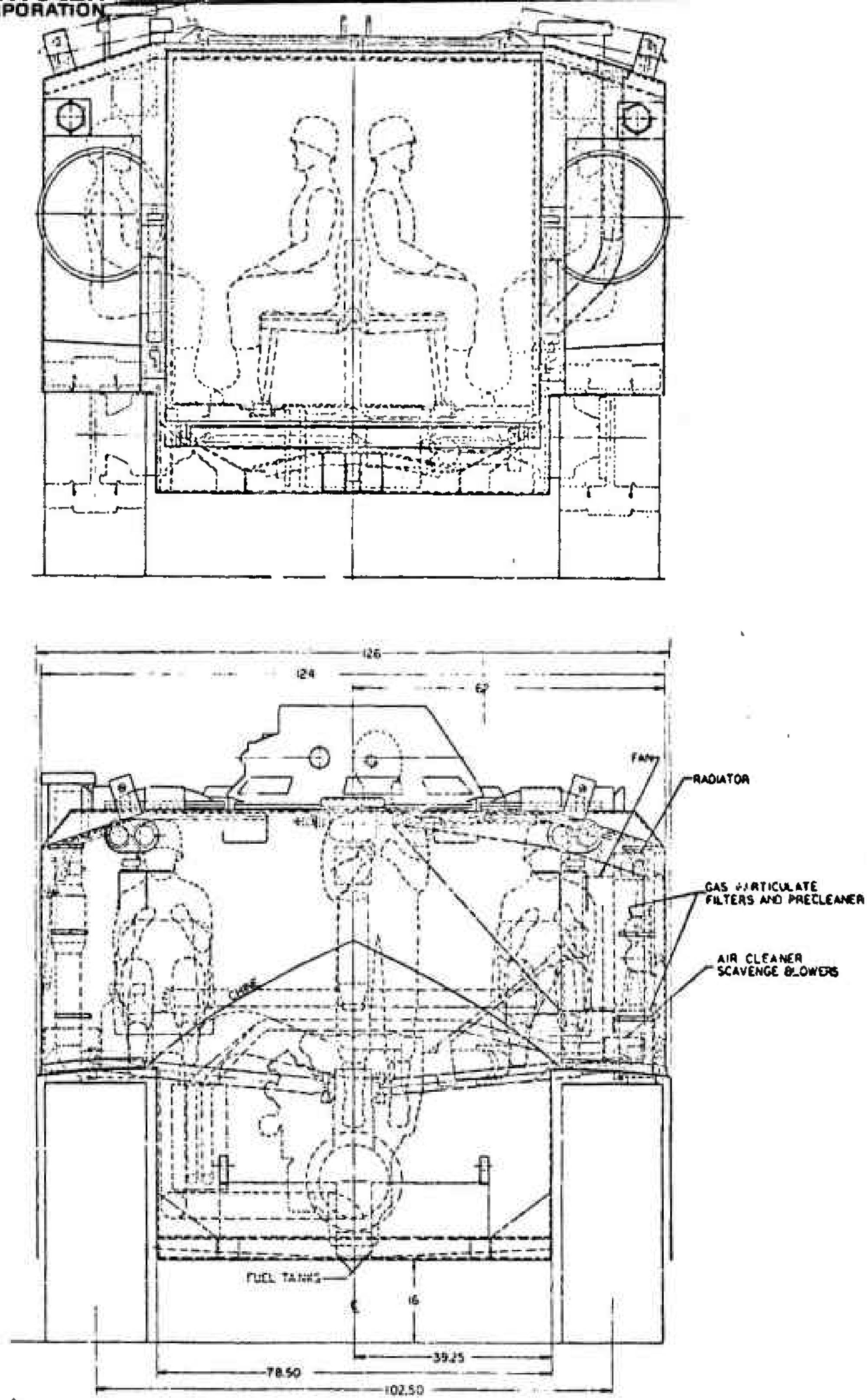


Figure 5-18 Screw Propelled Vehicle - End Views

turbine-bladed track grousers are retained, so that even without the bow fenders, a speed of 6 MPH or better is possible on tracks alone. A bonus of reliability is thus provided in case of damage to the propellers great enough to render them inoperative. Since the propellers extend beyond the sides of the vehicle only 28 inches and remain clear of the ground at all times, the possibility of damage must be acknowledged, though its probability be remote. When the propellers are housed, they will be partially submerged and will be able to move the vehicle about, but high speeds would be impossible under these circumstances. The turbine-bladed tracks enable the LVTPX12 to make a good speed under any possible conditions. Their presence requires the propellers to absorb some 40 more horsepower at 10 MPH, but their elimination would not reduce the required power down to the next suitable engine. Therefore, the screw-propelled LVTPX12 employs the same engine as the track-propelled version.

While the necessary power for 10 MPH by the twin screws is not quite so high, no advantage in weight can be gained by installing an engine of lower power than the GM-12V71T (See Figure 8-11 in Section 8.0). It is therefore proper to install the GM-12V71T, and thus, attain speeds even greater than 10 MPH by screws, or speeds of 6 MPH by tracks, at the option of the driver.

The Chrysler LVTPX12, capable of speeds over 10 MPH, is succinctly described in the tables of characteristics, Paragraph 5.12.3, where the particulars of the final design are compared with the specifications.

5.12.3 Tables of Characteristics. In Figure 5-19, the physical characteristics of both the track-propelled and the screw-propelled LVTPX12 are listed beside the specifications. The waterline length is not included in this list

	Specification	Track-Propelled Vehicle	Auxiliary-Propelled Vehicle
Weight (combat loaded)	45,000	51,990	53,670
Weight (less crew, fuel, stowage)		37,350	40,480
Cargo Capacity	10,000	10,000	10,000
Troop Capacity	25	25	25
Unit Ground Pressure			
w/cargo	-	7.27	7.57
w/troops	-	6.64	6.87
Tread Width (C. to C.)	-	102.5	102.5
Tread Length	-	178.75	178.75
Length Overall	26'0"	26'0"	26'0"
Width Overall	10'6"	10'6"	10'6"
Height Overall	8'6"	8'6"	8'6"
Angle of Approach	35° Req. 65° Des.	55°	55°
Angle of Departure	35° Req. 45° Des.	86°	86°
Ground Clearance	16"	16"	16"
Cargo Compartment			
Length	14'0"	14'5"	14'5"
Width	6'0"	6'0"	6'0"
Height	5'6"	5'6"	5'6"
Floor Area	-	86.5 sq. ft.	86.5 sq. ft.
Volume	-	475.75 $ft.^3$	475.75 $ft.^3$
Cargo Hatch			
Length	6' min. 9' des.	8'4"	8'4"
Width	5'	5'	5'
Ramp Opening			
Height	5'6" min. 6' des.	5'6"	5'6"
Width	6'	6'	6'
Cargo Handling System	Rollers in Floor	Rollers in Floor and Ramp	Rollers in Floor and Ramp

Figure 5-19 LVTPX12 Physical Characteristics

because it varies with displacement, but in the curves of form of Section 4.0, the waterline length, the draft, and all other significant particulars may readily be obtained. If the weight of the vehicle is given for any displacement, the waterline length is very close to 25 feet. The weights of both the screw-propelled and the track-propelled versions, which exceed the specified weights, are given detailed attention in the following pages.

Figure 5-20 shows in detail the performance characteristics of each version alongside the specified performance.

5.12.4 Weight Evaluation. In Appendix N are detailed weight summaries of five vehicle arrangements. The evaluation of these five has resulted from the evaluation of several armor materials, engines, and transmissions. Section 21.1, Parametric Evaluations, will review and tabulate the many vehicle arrangement permutations which were evaluated before the final five configurations were selected.

The five vehicles represent the basic concepts, track and auxiliary propelled, with variations of power plant installations in the auxiliary-propelled version.

The five vehicle concepts are as follows:

Concept 1 - Track propelled, 12V71T engine; height = 8.5 feet, length = 26 feet, and width = 10.5 feet.

Concept 2 - Auxiliary propelled, 12V71T engine; height = 8.5 feet, length = 26 feet, and width = 10.5 feet.

	Specification	Track-Propelled Vehicle	Auxiliary-Propelled Vehicle
Maximum Water Speed, Forward	8 mph req. 10 mph des.	8.1	10.7
Maximum Water Speed, Reverse	3.5	5.0	6.0
Maximum Land Speed, Forward	30	38	36
Minimum Land Speed, Forward	5	2	2
Maximum Land Speed, Reverse	8	16	16
Maximum Grade, Forward Slope	60%	70% +	70% +
Maximum Grade, Side Slope	60%	60%	60%
Speed on 60% Slope	2 mph	2.5 mph	2.5 mph
Maximum Trench Crossing	8' wide - 4' deep	9.5' wide - 4+ ft. deep	9.5' wide - 4+ ft. deep
Maximum Vertical Obstacle	3'	3' +	3' +
Plunging Surf Capability	10'	Reserve Buoyancy 80% of Displacement	Reserve Buoyancy 80% of Displacement
Stability - Righting Capability Roll to Port or Starboard	90°	110°	100°
Stability - Execute Turn on 60% Slope	90° turn @ 2.5mph	90° turn @ 2.5 mph	90° turn at 2.5 mph
Endurance - Water Land	7 hrs. @ 8mph 300 mi. @ 25 mph	7 hrs. @ 8mph 584 mi. @ 25 mph	12.4 hrs. @ 8 mph 300 mi. @ 25 mph
Climatic Operation °F	-25 to 125	-25 to 125	-25 to 125
Climatic Operation w/Kit °F	-65	-65	-65
Armament - 20 mm 7.62 mm	20mm	20mm & .30 cal.	20mm & .30 cal.
Armor Protection Artillery Fragmentation	99% of 105 mm Air burst at 50'	99% of 105 mm Air burst at 50'	99% of 105 mm Air burst at 50'
Rounds	5000 rounds equivalent of 7.62mm	325 rounds 20mm 700 rounds .30 cal.	325 rounds 20mm 700 rounds .30 cal.

Figure 5-20 LVTPX12 Performance Characteristics

Concept 3 - Auxiliary propelled, 8V53T engine; height = 8.5 feet, length = 26 feet, and width = 10.5 feet.

Concept 4 - Auxiliary propelled, 8V71T engine; height = 8.5 feet, length = 26 feet, and width = 10.5 feet.

Concept 5 - Auxiliary propelled, twin AC-350C engine; height = 8 feet, length = 26 feet, and width = 10.5 feet.

These concepts meet all of the desired vehicle performance requirements with one common exception: gross vehicle weight of 45,000 pounds. The gross weights vary from 51,990 pounds for the track-propelled concept to a maximum of 53,670 pounds for the auxiliary-propelled concepts. The following table shows the major weight items and the percentage of gross vehicle weight they constitute. Percentages for the auxiliary-propelled vehicle represent low to high percentage range for the four concepts.

	PERCENTAGE OF GVW	
ITEM	TRACK	AUXILIARY
Hull Group	15	14.6 to 15.1
Armor	24	24 to 24.9
Track and Suspension	17	16.4 to 17
Power Train	10.1	12.2 to 15
Payload	19.2	18.7 to 19.2

Armor constitutes the major weight item. Appendix B gives a complete evaluation of the armor situation, where it is concluded that steel offers the best solution for overall weight, cost and fabricability. Yet, the lightest steel armor has not been selected. Use of dual-hardness steel armor on the

sides of the vehicle can reduce the GWW by 1090 pounds, but in Appendix B, it is concluded that the price paid for this weight savings is not justifiable. The best aluminum armor available offers no distinct weight saving as compared to the selected steel armor. The armor weights shown do meet the ballistics specifications. Reduction in this weight can be achieved only by using costly materials or by reducing the armor protection requirements.

The second largest weight item is the payload, which at 10,000 pounds, represents a fixed weight required by the specifications. The weight of track and suspension is the next largest item. The 16.4 to 17 percent of GWW actually represents a very lightweight, yet reliable system. Section 9.0 shows that no existing armored track-laying vehicle in the military system has a track and suspension weight of less than 20 percent of GWW.

The hull group generally constitutes 15 percent of the GWW. Although the basic hull structure is steel, liberal use of aluminum has been made in areas which can be designed to be completely independent of the basic hull structure and armor. Weight evaluation in Appendix B shows that weight can be saved by use of an aluminum structure but this weight saving is offset by the increased armor weight of aluminum, resulting in a basic armor-structure weight approximately equal to that of the selected steel armor structure.

The power train of the track-propelled concept is 10.1 percent of the GWW as compared to 12.2 to 15 percent of GWW of the auxiliary-propelled concepts. Full evaluation of the selections is in Section 8.0. The selection is predicated on water performance and the resulting power requirements. Possible machinery arrangements of the final drive and transmission, compatible with

the engine power, is also a consideration, which means that weight is not the dominant factor in the selection of power train requirements.

The above items represent over 85 percent of the GVW and each, in itself, has a direct influence on GVW. The other items constituting the GVW do not have a large influence and would essentially be the same precentage of the total regardless of the basic design.

5.13 Description of Components of the LVTPX12. In addition to the general layout and the numerical description of the LVTPX12 in terms of its performance and physical characteristics given in the foregoing paragraphs, details of the important components are given below. Many of these components, armor for example, are identical for both the track-propelled and the screw-propelled version of the LVTPX12, and in such a case no distinction will be made between them.

5.13.1 Human Engineering. For optimum visibility on land, the driver is positioned as far forward as practicable on the port side of the vehicle. The assistant driver or the embarked troop commander occupies a symmetrical position on the starboard side. For maximum effectiveness, the automatic weapon cupola is located forward on the centerline between the driver and assistant-driver positions. Interior arrangements provide for complete freedom of access between crew stations and troop compartment and permit positioning the gunner so that he may operate his weapon in 360 degrees azimuth from a standing position.

Embarked troops are seated on benches along the track channel and upon removable seats down the longitudinal center of the troop compartment.

These seats are folded for stowage on the track channels when not required. Fully adjustable seats are provided for the crew to permit use in the elevated position when the vehicle is operating ashore with cupola hatches open.

Careful study has been given to rapid and orderly exit of both passengers and crew in operations and in emergencies.

The design of the ventilation system has been with emphasis on maintaining the battle fitness of personnel. The engine exhaust is downward, away from intake ducts. Special ventilation in the gunner's cupola insures his maximum efficiency.

5.13.2 Hull and Structure. Both the track-propelled and the screw-propelled versions of the amphibious personnel and cargo carrier are constructed of integral steel armor. The boat bow, designed for maximum efficiency in water speed, also reduces to a minimum the amount of flat area presented to oncoming projectiles from any direction. The bow has been designed with a developable surface to avoid furnacing of plates. The deck height of 8.5 feet has been retained as protection against swamping at maximum possible water speed, 10.7 MPH.

A large hatch in the top deck expedites handling of cargo. In addition to the rear ramp, the driver's and assistant-driver's cupola hatches forward and a top access hatch abaft the cargo hatch provide passage for personnel. Emergency egress is through escape hatches on each side of the troop compartment.

The armor plate acts as the stressed skin of the structure in distributing the primary loads. Three main frames are supplemented by intermediates. In all framing, standard shapes of steels that combine high strength with easy weldability and economy are used as much as possible.

5.13.3 Armor. For the specified protection at least weight, lowest cost, and for serving simultaneously as the watertight skin, the strength web, and the protective shield, every possible material has been examined in Section 6.0 and Appendix B. The evaluation of these materials has included their maintainability, availability, corrosion resistance, sensitivity to temperature, and ease of fabrication. Based on these investigations, the armor selected for the LVTPX12 is:

- Top – MIL-Spec. steel plus nylon back
- Sides – BHN-500 steel
- Bottom – MIL-Spec. steel

5.13.4 Power Train. Nearly a hundred engines of various types and power ratings meeting the compression ignition or multi-fuel requirements were investigated. These included current production and developmental models of diesels, gas turbines, rotary combustion engines, etc., available within the time frame of the LVTPX12 program. Several promising units are described in Section 8.0 from which alternative choices may be made. The problem of availability of transmissions suitable for use at the power levels required for the high water speed targets was found to be much more critical. No military or commercial transmissions in production or advanced development are usable for the LVTPX12 without modification of case geometry or internal component design.

To provide the installed horsepower required for water speed, an 800 horsepower GM-12V71T Diesel engine is located in the forward part of the vehicle. A flat configuration CCS-1 countershaft-type transmission designed for the track-propelled LVTPX12 and a CCS-2 for the screw-propelled LVTPX12, is located beneath the floor at the stern of the vehicle and is connected to the engine by a drive shaft beneath the floor of the troop compartment. The transmission provides four (4) speeds forward and two (2) speeds reverse. The output of the transmission is close-coupled to the planetary-reduction final drives, which also include the modulated steering function. Complete details of the power train and associated systems are presented in Section 8.0.

The power train components and arrangement for the screw-propelled LVTPX12 are generally as described for the track-propelled vehicle. However, due to the addition of the laterally retractable outboard propellers, a modified drive line is required. Mounted on the rear of the engine is the auxiliary drive adapter gear box described in Section 8.11. This gear box provides power take-off to drive two shafts; one running atop each track channel to supply power to the retractable propeller assemblies.

5.15.5 Track and Suspension. The vehicle is supported on 12 rubber-tired road wheels (six per side) suspended upon torsilastic springs cantilevered from the sides of the vehicle. The rear-drive sprockets provide power to the side grouser lubricated single-pin amphibious track, which is fitted with removable rubber-tread pads permitting operation ashore without damage to improved roads. A compensating front idler is provided to take up track slack during rough terrain operation and to allow adjustment for track pin and bushing wear. Return idlers optimize water propulsive efficiency and prevent

high dynamic loads inherent in flat-track return systems. A removable track shroud assembly encloses the track return assembly and controls water flow within the suspension envelope. Detailed description of the track and suspension system is presented in Section 9.0.

An amphibious type of side grouser track, identical to that used on the track-propelled version, is also proposed for the screw propelled vehicle. The track remains stationary during propeller operation and model tests have offered assurance that the added hull resistance due to this condition does not seriously increase hull resistance at the operating speeds. The horsepower provided by the 12V71T diesel engine is adequate to overcome the added resistance and provide sufficient power to drive the vehicle at the predicted speed of 10.7 MPH. Use of this track permits complete flexibility of operation. With propellers retracted, regardless of the load condition and consequent water line position, full tracked mobility and maneuverability is provided by engaging the track drive. Thus, in confined quarters where the outboard extended propellers would be vulnerable to damage, they may be retracted, with satisfactory propulsion then being provided by the track system. This redundancy in propulsive systems is a further advantage as a safety back-up system should the propeller system become inoperative during water operation.

5.13.6 <u>Appendages.</u> In order to provide maximum water propulsive effort, two appendages to the track and suspension to hull interface are provided: namely, a wrap-around front fender surrounding the front idler assembly and a wrap-around rear fender with a short contravane attached which surrounds the rear sprocket and track envelope. Both fenders are retracted by rotation to provide adequate clearance for negotiation of vertical obstacles and for

permitting full capability of the angles of approach and departure designed into the hull and suspension assemblies. The effects of these appendages upon the overall water performance of the vehicle are fully described for the screw propelled vehicle, in Section 4.0. Because maximum propulsive efficiency is not required of the track system for the screw propelled vehicle, the fenders and contravane can be eliminated.

5.13.7 Auxiliary Propulsive Devices. Primary water propulsive effort is provided by the two laterally retractable propellers mounted at each side at the rear of the vehicle. The geometry of the mounting struts is designed for maximum possible depth of immersion of the propellers when in the extended position without undue vulnerability to grounding in the surf zone. Power can be supplied to the propellers at any position throughout the retraction arc. Location of the propellers in the stowed position is such that in the light load condition most of the propeller is out of the water. Thus propulsive thrust will be extremely low requiring the use of the tracks for maneuverability. Since both tracks and propellers can be operated simultaneously, no complications are involved in driving procedure.

Excellent steering characteristics of the vehicle under propeller drive are predicted due to the use of controllable pitch propellers. These units are independently controllable by the driver from full ahead pitch, through neutral to full reverse pitch. Thus a reversing gear box or transmission is not required with this installation. Varying the relative pitch between the two propellers will provide differential thrust at a wide lever arm furnishing an unusually large turning moment for a craft of this size. The pitch differential will permit the driver to maintain a straighter course with minimum steering effort.

There is nothing novel about controllable pitch propellers. For both aircraft and marine use, they have been perfected for many years. Their reliability and utility is proven about as well as any mechanism could be. Application of the controllable pitch principle to the difficult steering problem of the LVTPX12, however, is a new idea.

As shown in Figure 5-17 (see also photographs of the model in Section 4.0) the propulsive units of the LVTPX12 swing directly outboard, port and starboard, to their extreme positions above the groundline but well below the load waterline. Thus, they are not vulnerable in landing. In fact, no harm would come to the propellers after landing if the driver failed to retract them, unless he passed closer than 28 inches to an obstacle. In this extreme position, the two propellers are capable of a maximum turning moment of 25,000 pounds-feet, but even with such a turning moment the propellers exert no serious heeling or trimming moments on the vehicle.

The pitch (distance of advance per revolution) of the propeller is controlled by controlling the angle of the blades. This is done by movement of a cam within the propeller hub. Actuation of the pitch control mechanism may be through mechanical, electrical, or hydraulic linkage from the operating station, or by a combination of these. The response of pitch to signal from the operator is reported back to the operator by a follow-up or telltale system.

In all the several controllable pitch systems on the market, control of pitch is from the operating station (the wheelhouse on ships) by movement of a small lever. Depending on the size of the propeller and the response time desired by the designer, the pitch of the propeller may be changed from full ahead to

full astern in as little as 2 seconds. A response time of 5 seconds from full ahead to full astern has been judged sufficient for the LVTPX12, but this time can be reduced if it is deemed necessary. This time is not a measure of the time required for small changes in pitch for steering. The response of the vehicle to small course corrections in steering would be very nearly that of a boat of equal moment of inertia, but much more positive in sharp turns than could be achieved by rudder.

The propulsive units are raised and lowered by hydraulic rotary actuators at the will of the operator. The time for movement from one extreme to the other is 4 seconds. This time is selected because it is adequate for maneuvering and yet does not require great hydraulic power. The time can easily be shortened by installing more expensive, larger, and heavier hydraulic machinery. The nozzle of the propeller clears the ground 27 inches in the outboard position. In the following table several geometric characteristics of the propulsive units are provided: column (2) shows the angle to which the vehicle must roll to raise one propeller clear of the water. Column (3) shows the percent to which the housed propeller is immersed under various conditions of loading. Column (4) shows the depth of the top of the nozzle below the still water line for these conditions of loading.

(1) Condition	(2) Angle of roll - degrees	(3) Percent immersion	(4) inches below still water
Light	22	0	8
Troops	31.5	60	21
Full	36	100	30

In colume (I) "light" means full fuel and crew only; "troops" means loaded with troops, using a round figure of 5,000 pounds for their weight; "full" means loaded with 10,000 pounds of cargo, full fuel and crew.

The depth of the propellers below still water in the outboard condition is not the depth when underway at full speed, since the stern wave is above the screws. Although the retracted propellers will push the vehicle when immersed or only partly immersed, the resulting speed will not be very great because of the poor ability of water to get to the screws. (See Appendix A, Section 24.0) In the extreme light condition the propellers when housed will be mostly above the water line so that in some cases the tracks will have to be depended on for propulsion and maneuvering. If operating conditions are such that the propellers can be lowered, they will be immersed adequately for full speed under any load condition. The smallest rolling angle which will bring one propeller clear of the water is 22 degrees. This will require severe rolling. It is shown in Section 4.0 that the LVTPX12 will not get into synchronized rolling except in small waves.

5.13.8 Armament. The forward turret is the same in both the track propelled and screw propelled concepts. It mounts a 20 mm Hispano Suiza gun and a co-axial .30 cal machine gun. Sighting of the weapons is accomplished through a bore sighted monocular periscope and a wide angle, direct view vision block mounted side-by-side in the gun cradle. Control of both weapons in azimuth and elevation is power assisted. Both weapons are supplied with ammunition through feed chutes from conveniently positioned storage boxes. Ejection of the 20 mm spent brass is overboard through an ejection port. .30 cal spent brass is collected inside the turret.

An exhaust shield-blower arrangement eliminates fumes during firing. A top access hatch in the turret provides for the gunner's convenience in observation and servicing of the weapons. A complete description and analysis of the turret and its components is provided in Section 16.0.

For convenience of small arms support fire by embarked troops vision periscopes and firing ports are provided in the sides of the troop compartment.

5.14 Alternatives. The track propelled and screw propelled versions of the LVTPX12 described in Paragraph 5.12 meet or exceed the specifications in every respect except weight. In addition to specifications and engineering analyses, economics and military science must also supply the bases for judgment. The specifications in several areas are not numerically fixed, but are described within brackets of maxima and minima. Some specifications, an example being maneuverability, are not numerically described. Therefore it is proper to consider the effect of varying some of the salient characteristics of the LVTPX12.

The following paragraphs contain information on the effects of selected alternatives that might be judged significant by military evaluators. The facts supplied here are confined to the engineering of the LVTPX12. The application of these facts to military operations, and decisions on their economic consequences, are best performed by those skilled in other disciplines. The following subjects will be reviewed:

- Height
- Means of Reducing Weight
- Reduction in Width

- Effect of Reducing Length
- Appendages for Track Propulsion

5.14.1 <u>Height.</u> From the standpoint of military tactics, a low silhouette is much to be desired. The specified 8 feet 6 inches is an upper limit, but the desired height is as low as may be feasible. It has been shown that inside headroom could be decreased as much as one foot and still accommodate troops or the small wheeled vehicles expected to be transported in the LVTPX12. It is shown in Section 4.0, however, that the freeboard cannot be decreased if the speed of 10 MPH is to be maintained. The tests on bow wave height were in calm water. In waves of moderate height the water will begin coming over the bow. The track propelled LVTPX12 can stand a reduction of 8 inches of freeboard for absolutely smooth water. In a moderate sea, however, the vehicle would not be able to maintain 8 MPH if the freeboard were reduced. Furthermore, reduction of deck height will decrease the reserve buoyancy, decrease the righting arm at large angles of heel, and consequently reduce the ability of the vehicle to negotiate heavy surf. The weight will be reduced approximately 60 pounds for each inch of reduction in height. The feasible height of deck can be reduced in three general ways:

- Reduce water speed.
- Reduce weight.
- Increase stern trim.

On the average, a weight of 1200 pounds immerses the craft and reduces freeboard one inch. Therefore each 1200 pounds of reduced weight allows one inch to be cut off deck height, while allowable speed will not be affected. Acceptance of significantly reduced water speed will combine all three of these

factors, since a smaller engine will take weight out of the bow, increase stern trim, and reduce GVW. The possible methods of reducing weight, and their consequences, are reviewed in the next paragraph, and the possibilities of reducing power follow in Paragraph 5.14.3.

5.14.2 <u>Means of Reducing Weight.</u> Since the early stages of this study it has been apparent that the desired 45,000 pound GVW cannot be achieved unless:

- The desired requirements be compromised, i.e., armor protection, cargo capacity, water speed.
- An increased cost be accepted.

Paragraph 5.12 of this section shows the items constituting the major part of the GVW. The largest single weight item is armor and it was explained that no additional weight saving can be achieved unless the specifications be compromised or additional cost accepted. The parametric study tabulations, Section 21. , show several configurations of about 45,000 GVW. The armor system used in these consists of a combination of titanium, nylon and boron carbide ceramic-aluminum armor. Obviously the material and fabrication costs of this armor system are very high compared to conventional steel or aluminum armor. There is also the added complexity of fabrication in joining two dissimilar metals. Mechanical joints are not preferred for ballistics, and are not satisfactory for maintaining watertight seals.

The second largest weight item is cargo capacity, fixed at 10,000 pounds by the specifications. Cutting this by 50 percent will not reduce any of the five acceptable concepts, Paragraph 5.12, to 45,000 GVW. Reduction of the hull envelope sizes will reduce weight but minimize the probability of a 10,000 pound cargo capacity.

Another major weight item is the hull structure, excluding armor. Structural weights are directly dictated by material strength and loading conditions. Efficient sections commensurate with loads, material strength, and interior limitations have been designed. It was previously stated that an aluminum structure results in less weight than a steel structure but this weight is compensated by the increased armor weight of aluminum. Reduction in hull envelope can reduce structural weight, because less material is required. The parametric studies show that this reduced weight is not of significant magnitude except for a narrow vehicle. It should be noted, however, that approximately half of the hull weight is not influenced directly by armor selection, hull envelope or primary structural loadings.

Suspension weight is directly influenced by GVW. A very lightweight suspension system has been designed, but no reduction in suspension weight is possible unless reliability is sacrificed or the GVW is reduced to 45,000 pounds.

It was stated in Paragraph 5.12 of this section that variations of power train components do not produce a large effect on weight. Nevertheless, if the horsepower requirements were reduced so that the 8V71T engine could be substituted for the 12V71T, more than 700 pounds could be saved. A further reduction in required power could save more than 1500 pounds by substituting the 6V53T engine for the 12V71T. Meeting the land performance requirements with the 6V53T engine at 45,000 GVW would be marginal. (See Paragraph 5.10.3)

All of the five vehicle concepts (Paragraph 5.11) meet all of the specifications except weight. To achieve the desired GVW may mean sacrificing some other requirement. The parametric study shown in Figure 5-21, however, shows a track propelled configuration which meets all of the specifications. This

VARIABLE	DESCRIPTION	WEIGHT	REMARKS
Propulsion	Track	-	NARROW VEHICLE
Water Speed	8 MPH	-	WIDTH - 8 FT. 8 IN.
Length	26 FT.	-	VEHICLE MEETS ALL SPECIFICATION
Height	8 FT. 6 IN.	-	REQUIREMENTS
OHP	602	-	
GHP	800	-	
Engine	(2) KC4-90	900	
Transmission	CCS-1	800	
Power Train	AND ENG. ACCESSORIES	1410	
Auxiliary Drive			
Systems Weights			
Electrical		785	
CO_2 System		110	
Hydraulic		575	
OEM		1040	
Crew		600	
Armor	FULL PROTECTION	8612	
Top	1/2"TITANIUM + 3/4"NYLON	-	
Sides	1/4"B_4C CERAMIC+3/8"ALUM.	-	
Bottom	7/16" TITANIUM	-	
Fuel	AND SYSTEM	3050	
Structure		6885	
Payload			
Troops		-	
Cargo		10,000	
Armament		1475	
Suspension		8600	
Total Weight		44,842	

Figure 5-21 Description of a Vehicle Meeting All Specification Requirements

concept uses an engine that is not fully developed, an armor system that is expensive and difficult to fabricate, and a track and suspension that would be difficult to build within the hull confines. The water speed is estimated and not based on actual model tests, but it was assumed that the resistance of the resulting hull would be the same as the optimum hull designed from the testing program.

The steering characteristics are subject to the same objection as that of the narrow vehicle described in Paragraph 5.5.2. While from the standpoint of the comfort of crew and passengers it might be possible to accept less softness of ride, and thereby allow a shorter wheel travel, the prime reason for the soft suspension system is not altogether the comfort of personnel but rather the damping of the pitching motion of the vehicle in fast cross country travel. If a lower land speed can be accepted, of course, the wheel travel can be reduced, the wheels enlarged, and consequently fewer wheels will be required.

Thus, this lightweight vehicle will not only be costly, but concessions in either quality of performance would in all likelihood have to be made if the prototype were built. Similar analyses have been made for the other concept variations summarized in the parametric study forms in Section 21.0.

5.14.3 Possibilities of Reducing Engine Power. Engine power can be reduced if a consequent decrease in water speed is acceptable. A 300 horsepower engine is the smallest engine that could be considered for land propulsion and this only with the most efficient transmission. With outboard propellers, however, this engine could drive the LVTPX12 at 8 miles per hour. Reduced engine power saves weight in fuel, cooling system, drive line, etc., but a small reduction

in power will not save very much weight in the engine. The GM 12V71T engine is no heavier at 800 horsepower than the lightest, available engine at 600 horsepower (See Figure 8-1) and Paragraph 8.2 in Section 8.0) Section 5.0 in Appendix D contains data on several combinations of engines and transmissions and their effect on vehicle speed. Figure 5-22 is a summary of some of the pertinent combinations from Appendix D illustrating the effect that reduced engine power has on vehicle weight and performance.

5.14.4 Reduction in Width. A narrower vehicle has already been shown to have no significant advantages in water speed, although the probability is that less beam would improve performance in head seas. The range of stability is not seriously diminished, the narrow vehicle still having a very ample righting arm at 90 degrees of heel. The moment to trim one inch is 4 percent less than for the full width. These investigations were for a reduction of 20 percent, and any smaller reduction would have corresponding effect. In the version investigated, the tracks were crowded into a height of 32 inches in order to maintain the inside clearance of 6 feet. If this inside clearance were sacrificed, of course, the height for tracks would not have to be lowered. This version would still be subject to the same unfavorable steering ratio, however, even though it would be able to go between closer obstacles.

For estimating purposes, reducing the width of the LVTPX12 by one foot would result in a reduction of weight of approximately 1,000 pounds, assuming no changes in other major characteristics.

5.14.5 Effect of Reducing Length. The first result of reducing length would be to increase the required power to maintain a given water speed. For example, the power to push the track propelled vehicle 8 MPH would increase 5 percent if

PROPULSION METHOD	ENGINE GROSS POWER	VEHICLE WATER SPEED	DRIVE TRAIN* DRY WEIGHT	FUEL AND FUEL SYSTEM WEIGHT**	TOTAL WEIGHT
TRACK	1120	8.4	4715	3580	8295
TRACK	800	8.1	4092	3100	7192
TRACK	720	7.9	4265	2750	7015
TRACK	530	7.2	3453	2360	5813
PROPELLER	1120	11.0	7405	1565	8970
PROPELLER	800	10.7	6587	1664	8251
PROPELLER	720	10.5	6910	1495	8405
PROPELLER	530	9.0	5878	1440	7318
PROPELLER	400	9.0	5532	1385	6917
PROPELLER	300	8.2	5112	1555	6667

* ENGINE, PROP SHAFTS, TRANSMISSION, FINAL DRIVES FOR TRACK PROPELLED VEHICLE; PROPELLER UNITS AND DRIVES ADDED FOR AUXILIARY PROPELLED VEHICLE.

** TRACK PROPELLED VEHICLE FUEL CONSUMPTION CALCULATED FOR WATER MODE AND AUXILIARY PROPELLED VEHICLE FUEL CONSUMPTION CALCULATED FOR LAND MODE.

Figure 5-22 Engine Power Levels

the length were reduced one foot, the displacement remaining the same. The trimming moments of craft of different lengths, but in all other respects the same, are as the ratio of the squares of their lengths. A decrease in length would require that a sacrifice be made in fineness of bow, size of engine, or length of cargo compartment. At the same time, if weight is not reduced concurrently with length, the ground pressure will rise above the present maximum of 7.57 psi. and the steering ratio will decrease. The natural pitching

period of the vehicle will be reduced both on land and water, and the ability to travel across country at high speeds will be reduced.

5.14.6 Appendages for Track Propulsion. For the LVTPX12 equipped with twin screws, the bow fenders and stern contravanes have already been omitted because they are not needed for maximum speed by propellers. By track propulsion, the price of omitting them is a reduction in maximum speed to 6 MPH while the reduction in gross vehicle weight is negligible.

5.15 Conclusions. The arrangement of an amphibious vehicle is shown to be a complex exercise in engineering analysis, trade-off evaluations, tactical and operational judgment. Restraints are generated by the relative importance assigned to the many possible features of such a vehicle. The designer's first difficult problem, therefore, is to select the major controlling features before beginning the engineering details.

Chrysler's primary effort has resulted in the definition of two basic arrangements for amphibious vehicles, one track propelled and one auxiliary propelled, based upon the following order of priorities:

- Maximum specified water speeds
- Maximum specified ballistic protection
- Maximum specified exterior dimensions
- Mimimum vehicle weights

These controlling factors have led to the general arrangements depicted on the frontispieces of this Technical Study and described in general in Section 5.0 and in detail throughout the study.

To provide for a complete coverage of the many problems involved, many other configurations, arrangements and ideas were explored, some of which are briefly discussed in the foregoing paragraphs. Results of the arrangement studies show that the decisions reached are for the optimum arrangements to meet the specified requirements. Should any of the basic characteristics specified be significantly altered, or should the order of priority be changed, a wide variety of arrangements could result. Each variation would dictate some compromises in desired features, some advantages and some disadvantages. No one configuration is best in all areas. The recommended concepts come closest to satisfying the basic requirements and represent the optimum arrangements of features, components and systems for the LVTPX12.

SECTION 6.0
ANALYSIS OF ARMOR

6.0 ANALYSIS OF ARMOR

Because of its significant effect on vehicle weight and cost a major area for special study in the design of the LVTPX12 is armor protection. While the ability to stop specified projectiles under given conditions of attack remains the prime function of armor, it concurrently provides the watertight skin and serves as the chief strength member. Assuming equal satisfaction of the protective function, materials may then be compared on their secondary capabilities, but in addition they must be compared on their merits in fabrication, economics, maintainability and availability.

6.1 Security Classification. Rather than segregate the classified portion of the armor discussion, the contractor presents all armor discussion under Appendix B. The appendix is classified. It is a complete report on the armor study.

Appendix B will contain the study of armor and lead to conclusions for design of an optimum armor-structure system for the LVTPX12. The discussion is arranged in Appendix B as follows:

- LVTPX12 armor requirements
- Initial selection of acceptable materials
- Chrysler armor test program
- Evaluation of acceptable materials
- Conclusions

6.2 <u>Conclusions from Appendix B</u>. The conclusion of this engineering study is that the optimum armor material for use on the LVTPX12 is a combination of steel and steel-nylon. The combination would consist of:

sides - BHN 500 steel armor

top - MIL-S-12560 steel armor plus MIL-C-12369 nylon armor

bottom - MIL-S-12560 steel armor

The selection of steel for the primary armor makes possible a change to dual hardness steel armor for the side protection at any time up to and even during production if the accompanying weight reduction is deemed worth the additional cost. It is anticipated that further developments will reduce the current cost of dual hardness steel armor.

SECTION 7.0
HULL STRUCTURE

7.0 HULL STRUCTURE

Two important and significant parameters of this vehicle are its total weight and cost. The hull and associated structure for a tracked amphibious personnel carrier represent a large portion of the vehicle's total weight and cost. The weight enters into nearly every operational requirement and influences component selection. Hull configuration and structural arrangements are influenced by the specifications for the total vehicle and subsystems. For example, the requirements for ramp, cargo hatch, and access openings disrupt the structural continuity, so that additional framing members are needed to provide the required strength and stiffness. The effects of these and other requirements on the hull are discussed in other sections. The object herein is to define a hull to satisfy the explicit and implicit requirements with sufficient structural integrity, that will be lightweight, easy to fabricate, and low in cost.

The hull structural system for the LVTPX12 is the armored shell with a minimum amount of framing and is an example of semimonocoque construction. In the interest of economy of material fabrication and vehicle assembly, yet compatible with water operation criteria, the shell is constructed entirely of flat plate except for the developable curved surfaces at the bow. The effect of armor material selection on the hull structure is shown in Section 6.0.

Considering total cost effectiveness impact with regard to the cost, weight, strength and ballistic protection parameters, steel is selected as the armor/structural material. (See Section 6.0.) The preliminary hull designs for the LVTPX12, described later in this section, are based on this material for the armor plating and framing with aluminum used extensively for interior items.

7.1 Factors Defining Hull Structure Characteristics. In this section, the geometric and physical hull characteristics which define the boundaries of the structural design are examined. These characteristics result from the following vehicle requirements:

- Ballistic protection
- Hull openings
- Interior configuration
- Water operation
- Cross-country operation
- Transportability

7.1.1 Ballistic Protection. The thicknesses of the external hull plating are determined by ballistic protection requirements. These thicknesses are different for the many suitable armor materials, as defined in the armor section, but in all cases are greater than required in an optimized semi-monocoque structure for the expected hydrostatic and shear loads. Therefore, the analysis of plate-stiffener combination for least weight is restricted to stiffener size and location with the plate thickness remaining constant.

7.1.2 Hull Openings. The hull has many external openings, varying in size from approximately 36 sq. ft. for the ramp and 40 sq. ft. for the cargo loading hatch, down to less than 1 sq. ft. for air ventilation. These openings or cutouts are discontinuities in the structure and introduce local framing requirements along with restrictions in the main frame location. The ramp and loading hatch require substantial framing because such large openings detract significantly from the inherent strength and stiffness of the idealized closed box.

The numerous other smaller openings, such as personnel hatches and turret cut-out, require less local framing but their relative locations influence to a great extent the number and complexity of the main frames.

7.1.3 Interior Configuration. The features of the hull interior most pertinent to the structural design are the cargo and personnel requirements and the machinery arrangement. The inside cargo compartment dimensions of 14 ft. long by 6 ft. wide by 5-1/2 ft. high, coupled with a desired minimum exterior envelope, dictates the frame member depth restriction. This means that a compromise between many narrow frames versus few deeper ones and a possible larger hull envelope must be made. The space requirements for personnel, based on human engineering standards, have the same influence as the cargo dimensional criteria. The machinery arrangement affects both the spacing and size of frames. It is desirable that frames be close to these components for efficient mounting and yet be non-restrictive, so that convenient maintainability and removability is preserved.

7.1.4 Water Operation. High water speed and surfing operations impose unique requirements on the hull. The external form must have minimum resistance, but must agree with other configuration requirements. Open water operation imposes no significant hull bending or torsion, but surfing operations create high local loads.

7.1.5 Cross-Country Operation. Cross-country operation subjects the vehicle to large loads at low frequency. This loading environment is significant to the machinery, suspension and other component bracketry and mounting. It also imposes local design requirements for loads associated with crossing obstacles

and trenches. The racking and twisting resulting from uneven terrain establish qualitative criteria for the overall hull design as well as the framing requirements of the larger hull openings such as ramp and cargo hatch.

7.1.6 Transportability. Transportation by ship and rail are the primary methods of long distance movement for LVTPX12 vehicles. To facilitate the procedure of ship transportability, criteria are established in MIL-STD-209B, and are applicable to this vehicle. Design load conditions for the lifting provisions are defined in MIL-STD-209B. Shipment by rail imposes large g-loads along the vehicle's longitudinal axis due to "humping" conditions. This load is significant to the design of machinery and component mounts.

7.2 Design Load Conditions. The significant sources for loading conditions used in the design of the hull and associated structure are as follows:

1. Cross country operation resulting in vehicle accelerations and high local impact loads.
2. Static and dynamic water pressure loading.
3. Towing and hoisting.

These general categories establish guidelines for the determination of overall loading conditions and local design criteria. In most cases these criteria are quantified recognizing the absence of data relating actual operation to resulting stresses. Also the random variations in loading are nearly impossible to predict throughout the vehicles' expected life. Experienced engineering judgement obtained from association with similar vehicles such as tanks, bridge ferry, and armored cars is drawn upon to set up the loading criteria for suspension components, towing devices and areas subjected to local

loads from adverse terrain. The assumptions and calculations made in defining the load conditions and load factors are shown in Appendix C.

The design loads and load factors used are considered to be static equivalents with safety factors included. The magnitude of the safety factor is variable as influenced by the vagueness of the relationship between the design loads and those experienced in vehicle operation.

The materials' guaranteed minimum yield strengths are used as the allowable stress except where buckling is critical, in which case it is determined from equations of elastic stability. Where the materials' physical properties are reduced because of welding, the allowable stress is obtained from the manufacturer or determined from test data.

A summary of the load conditions and load factors used in the preliminary structural design is presented below.

1. Hull plating and main frames
   a. 1000 lb/ft^2 pressure on top deck.
   b. 500 lb/ft^2 pressure on sides, front, rear and bottom.
   c. 25,000 lb acting upward on any bottom frame.
2. Cargo floor
   a. Uniform distributed load of 413 lb/ft^2.
   b. Mighty Mite vehicle - moving wheel load and inertia load when secured - 1925 lb on 8" x 8", area.
3. Ramp (with one corner supported only)
   a. Mighty Mite moving wheel load - 1200 lb on 8" x 8" area.
   b. Uniformly distributed load of 236 lb/ft^2.

4. Lifting, towing, and mooring devices

<u>Lifting Eye</u>

a. Each eye loaded to .35% of shipping weight.

b. Maintain ultimate factor of safety of 5.

<u>Tow Eyes - Front</u>

22,500 lb each, acting within a pyramid $\pm$ 20 degrees from horizontal and $\pm$ 45 degrees from vertical plane.

<u>Pintle - Rear</u>

45,000 lb acting within a pyramid $\pm$ 20 degrees from horizontal and $\pm$ 45 degrees from vertical plane.

<u>Bitts</u>

12,000 lb in any direction in a horizontal plane.

5. Suspension

a. Road wheels - 8 g vertical.

b. Front idler - 1.5 GVW horizontal, 1.0 GVW vertical.

c. Drive sprocket - 1 GVW horizontal, .67 g vertical.

6. Overall restraint criteria

a. Horizontal $\pm$ 10 g.

b. Up 1.5 g.

c. Down 3.5 g.

d. Side $\pm$ 1.0 g.

7.3 <u>Structural Design Approach</u>. Preliminary structural designs have been performed for two vehicle concepts; one using tracks along for water propulsion and another using twin outboard propellers which retract into the aft hull sides. The vehicle characteristics and requirements which establish the

criteria of design as described in the previous sections have been considered in conjunction with the summarized loading conditions in fulfilling this task. The expected result of this or any preliminary design is to establish the feasibility of a concept to satisfy its intended function. Specifically, for the structural design of the hull this involves material studies, fastening methods, and structural member configurations and location. In this phase of the program less exacting techniques are used for determining the shape, size, and location of members and many miscellaneous brackets and supports are not even considered. This is justified because of the many interface details which become evident only during a detail design. Therefore, these preliminary designs will establish feasible structural concepts from which weight and cost can be determined but will be subject to further refinement in the detail design stage. A description of the preliminary design with definitive drawings is presented for the track propelled concept only. The auxiliary propelled concept is identical structurally in all respects except for the local details of mounting the outboard propellers. The assumptions and supporting design calculations are found in Appendix C.

7.3.1 Fabrication Techniques. Fabrication encompasses component manufacturing processes as well as the method of attachment for the total hull assembly. Rolled, extruded or preformed sections are best suited for framing members and structural supports and are used as much as practicable.

Welding is certainly the best method of permanent attachment for watertightness and also minimizes weight and cost. A weight saving results because overlapping flanges are not required. Also, welded joints can be made faster than mechanically fastened ones with less preparation and this results in cost savings.

In joining armor plates together, mechanical fastening is undesirable because the fastener may be broken loose and act as shrapnel inside the vehicle. Therefore, welding is the primary method of attachment that is used for the hull structure.

7.3.2 Material. The material used for the hull plating is steel. The basis for this selection is described in Analysis of Armor, Section 6.0. It is shown that for the required level of ballistic protection the steel armored hull can weigh less than one of aluminum and be less costly than hulls employing aluminum, titatium or ceramic for armor. There are several types of steel that can be used with their thicknesses varying from 0.25 to 0.31 on the sides, bow and stern; from 0.375 to 0.67 on the top; and 0.375 on the bottom. In the preliminary designs the thicknesses used are: top = 0.375, sides, bow and stern = 0.31, bottom = 0.375. The material used for framing members, gussets and structural supports that weld to the hull is high strength alloy steel, such as U. S. Steel's T-I and, low allow steel, cor-ten. These types of steel are gaining wide acceptance for use in large earth-moving equipment such as bulldozer and power shovel components where high strength and toughness at a reduced weight are desired. Their atmospheric corrosion resistance is superior to structural carbon steel by a factor of four to six times and cor-ten has been used in recent structural applications with bare exposed members. T-I's tensile yield strength is approximately three times standard structural steel while cor-ten's is approximately 1.5. Availability in sheet and plate form is good and a variety of standard rolled angles, channels and "I" beams can be obtained. Welding to any of the steel armor materials presents no problems with weld efficiencies approaching 100 percent.

Aluminum will be used for several interior items with the cargo floor, crew compartment floor, engine cover and cooling shroud being specific examples. The cargo floor with its roller system lends itself well to an extruded type similar to those used in large highway trailers. The other items mentioned are better suited for sheet and plate fabrication. Aluminum has a definite weight advantage over steel where panel buckling is the prime consideration and this criterion is applicable for the interior shrouds and baffles. In extruded form, heat treatable alloys of the 2000, 6000, and 7000 series are used since they have higher yield strengths than the non-heat treatable 5000 series alloys. For sheet and plate applications where extreme high strength is not required, alloy 6061 or 5052 will be used. They are the lowest priced of the weldable, structural alloys. The use of aluminum in a basically steel hull for the items previously mentioned presents no unique problems of fastening or electrolytic corrosion. Mechanical fastening would be used independent of material for component access and removal. Paint and cadmium or zinc plating will be used on component parts to eliminate any possible electrolytic corrosion between dissimilar metals. Allowable stresses for various materials is shown in Figure 7-1.

7.4 Basic Hull Structure. The hull structure consists of the armor plating which forms an irregular closed section stiffened and supported by transverse frames and intercostals. A stressed skin or semi-monocoque structural system is used because the vehicle requirements dictate a continuous armored shell, which when framed, provides the necessary strength and stiffeners for a minimum amount of added weight. The bow is boat shaped and is the only area of the hull where curved surfaces exist. These curved surfaces are developable and

ALLOY	MIN. TEN. YIELD PSI	TEN. ALLOW. PSI	COMP. (1) ALLOW. PSI	SHEAR (4) ALLOW. PSI
6061-T6 ALUMINUM SHEET & PLATE	35,000	35,000 (17,000) (2)	29,700 14,500	20,000 9,700
6061-T6 ALUMINUM EXTRUSION	35,000	35,000 17,000	29,700 14,500	20,000 9,700
5052-H34 ALUMINUM SHEET & PLATE	26,000	26,000 13,000	22,100 11,000	15,000 7,400
6062-T6 ALUMINUM EXTRUSION	35,000	35,000 17,000	29,700 14,500	20,000 9,700
5083-343 ALUMINUM SHEET & PLATE	39,000	39,000 21,000	33,000 18,000	22,000 9,700
6070-T6 ALUMINUM EXTRUSION	45,000	45,000 17,000	38,200 14,500	26,000 9,700
LOW-ALLOY STEEL (COR-TEN)	50,000	50,000	42,500	28,500
HIGH-STRENGTH ALLOY STEEL (T-1)	100,000	100,000	85,000	57,000
STEEL ARMOR PLATE (MIL-S-12560)	146,000 (3)	146,000	125,000	85,000
STEEL ARMOR PLATE (500 BHN)	225,000 (3)	225,000	190,000	125,000

(1) OR 75 PERCENT OF ULTIMATE BUCKLING OR COLUMN STRENGTH
(2) THE NUMBER BELOW THE LINE DENOTES AS WELDED ALLOWABLES
(3) ESTIMATED, 90% OF ULTIMATE TENSILE STRESS
(4) 57% OF MINIMUM TENSILE YIELD

Figure 7-1 Allowable Stresses

require no stretch forming. The radii of curvature are within the limits of the armor plating material. The functions of the frames, in a monocoque structure such as this, are to maintain its shape and distribute the applied loads to the exterior skin so that the entire hull cross section participates for overall bending and torsion conditions. The intercostals and intermediate members are provided to support components, introduce local loads to the hull, and beam loads between frames. The hull structural arrangement is shown in Figures 7-2 and 7-3. Three main transverse frames or bulkheads capable of distributing symmetrical and unsymmetrical loads exist and are the minimum number required to distribute loads so that the entire hull works as a unit. The bow, being continuous, serves as one bulkhead. Another is located at the juncture between cargo and crew compartment and the portal frame around the ramp is the third.

The cargo hatch opening has transverse frames located at each end and a partial one at mid-span. Longitudinal members stiffen its sides and butt into the end frames. In the crew-engine compartment transverse frames are spaced forward and aft of the turret. The fairly uniform frame spacing allows nearly equal distribution of external pressure loading. For the intermediate frames where this criterion applies, equal size and weight members result, at a given stress level. The external appendages, such as lifting eyes and towing devices, require local reinforcement. The lifting eyes are located at or very close to transverse frames and the front tow eyes are anchored to intercostals in the hull. A single towing device is provided at the rear, on the vehicle centerline below the ramp. It is a quick release type, operable from the top deck. A representative transverse frame is shown in Figure 7-4. Inverted angles welded to the hull form effective channel sections and are used exclusively

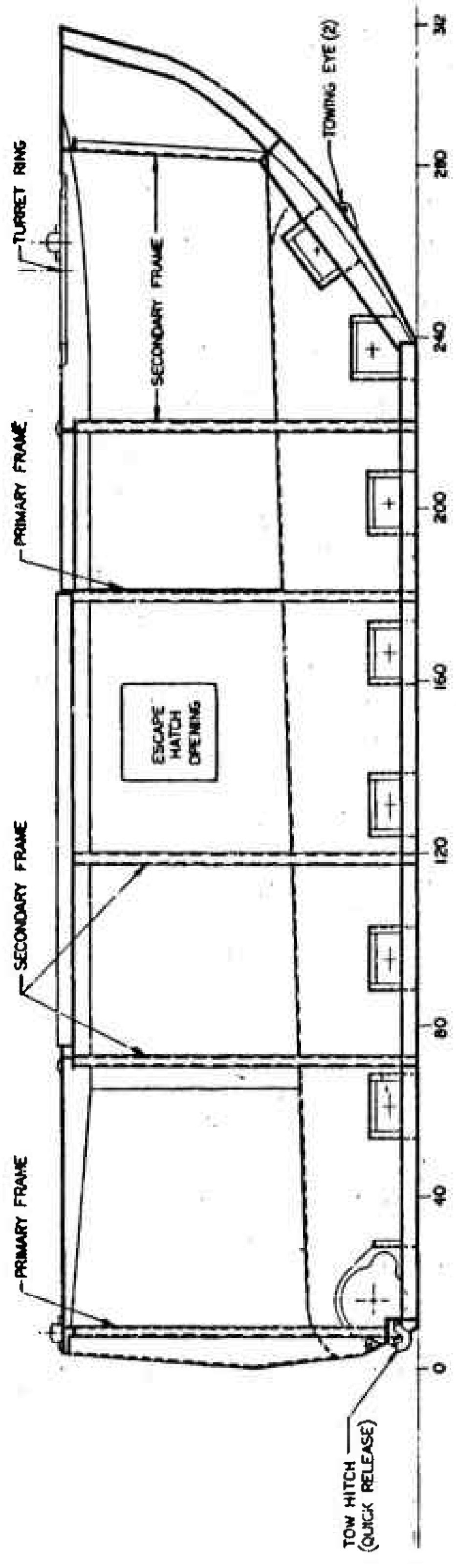


Figure 7-2 Structural Arrangement - Centerline Section

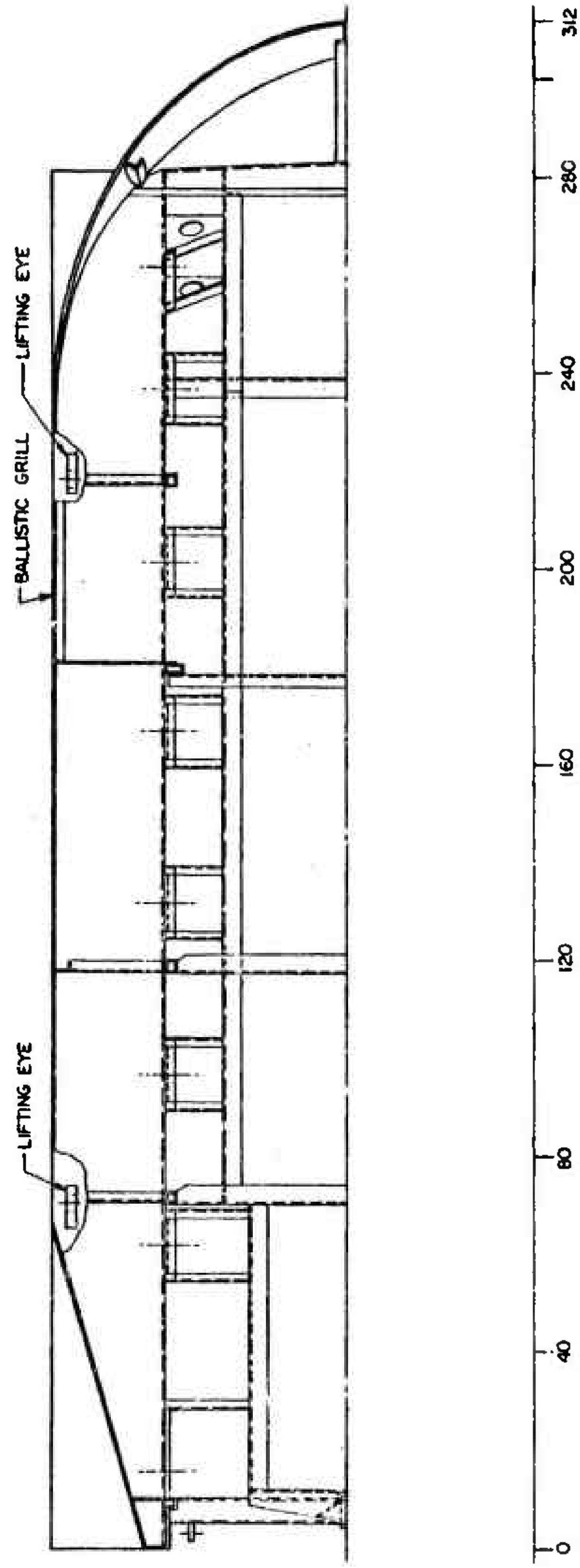


Figure 7-3 Structural Arrangement - Plan Section

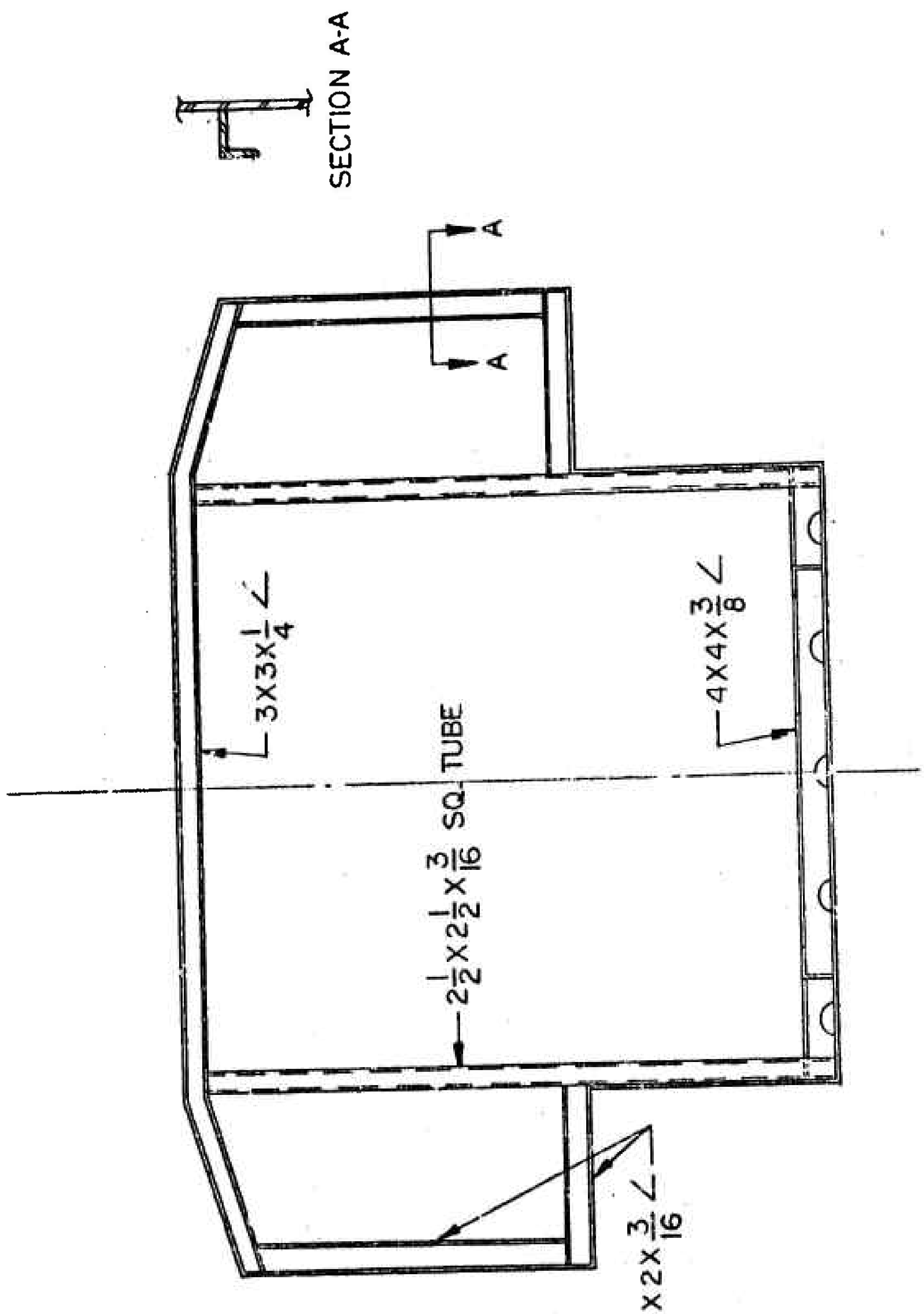


Figure 7-4 Transverse Frame

except for the vertical top deck supports which are rectangular tubes. These column supports serve two functions. Their primary purpose is to reduce the span of the deck members, allowing a shallower section. A secondary function is to resist the kick load that results from the change in direction of the deck member.

The suspension road wheels arms as well as drive and idler sprocket are cantilever mounted to the hull. The hull plating is backed up locally in these areas to distribute the cantilever moment to the hull. Typical support structure for a road wheel arm is shown in Figure 7-5.

The cantilever moment is beamed to the side plate and longitudinal member by the local transverse brackets. The longitudinal support is welded to the hull bottom and transverse framing members at its ends.

Boarding steps, located forward on the hull sides, are provided along with hand rails on the top deck. The personnel entrance hatch is located on the rear quarter of the top deck. Hatches over the driver and co-driver station are also provided. These hatches are primarily for allowing the vehicle operators to expose their head and shoulders but can also be used for egress and ingress. Two escape hatches are provided amidships, one on each side. These hatches are non-hinged and completely released by a single lever. Compression seals are provided around all hatch openings. Torsion bar or spring compensation is used on all hinged hatches except for the ramp.

7.4.1 Ramp. The ramp is mounted to the hull by two hinges and is opened and closed by hydraulic cylinders located on each side. Final securing of the ramp in the closed position is accomplished by a latch mechanism in the hull.

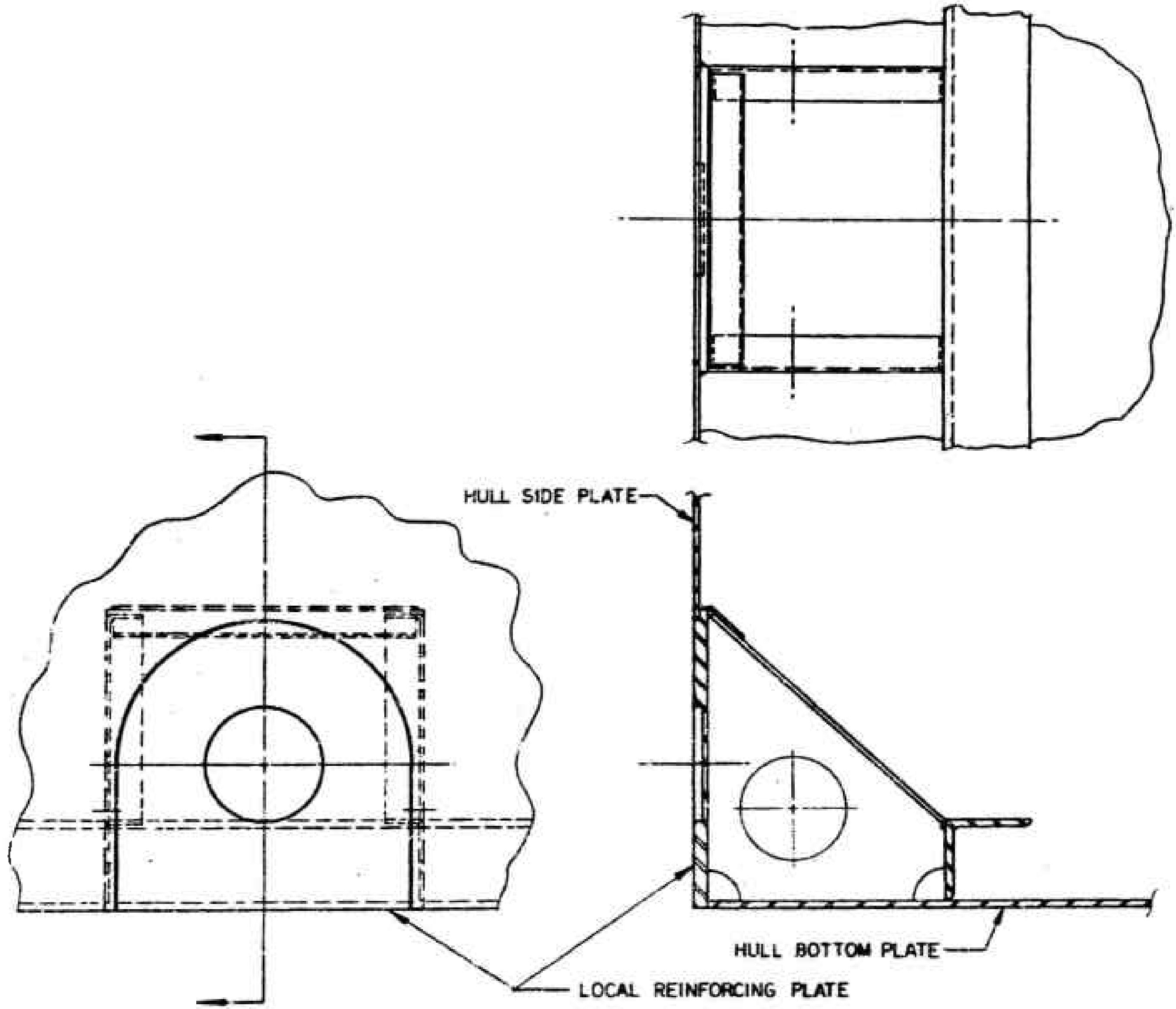


Figure 7-5 Suspension Road Wheel Support

A continuous seal around the ramp is provided and located in the hull. Details of the ramp operating mechanism are discussed in Sections 10.3 and 12.0.

When in the open position the ramp is an extension of the cargo floor. Because a cargo handling system consisting of rollers is located in the floor, they will also be incorporated into the ramp. The ramp is shown in Figure 7-6. The external surface of the ramp is .31 thick steel armor plate and provides the necessary ballistic protection. Enclosed sections are used to frame the edges and are welded to the armor plate. Closed framing members are used to develop the required torsional rigidity for unsymmetrical loadings, such as, when only one corner is supported in the open position. The extruded aluminum cargo floor sections with reversible floor rollers are used for the ramp's inner surface. This selection is made to save weight. The cargo floor extrusions provide an efficient deck surface. The decking is mechanically fastened to the welded ramp assembly with a sealer applied between the dissimilar metals to eliminate electrolytic corrosion.

7.4.2 Cargo Hatch Doors. The top deck cargo hatch closure consists of two rigid doors, of 3/8 armor plate with the edges reinforced by framing members. A transverse support is also used at the door's mid-span. To achieve the required level of overhead blast protection, a nylon blanket is used in conjunction with the armor plate. Four latches, two per door, are provided which are operable from the inside as well as top deck. Three hinges are used on each door with torsion bar compensation to facilitate opening and closing. The torsion bar is unstressed when the door is in a vertical position, so that its spring rate effectively reduces the force required to open and close it. The hatch closure is shown in Figure 7-7.

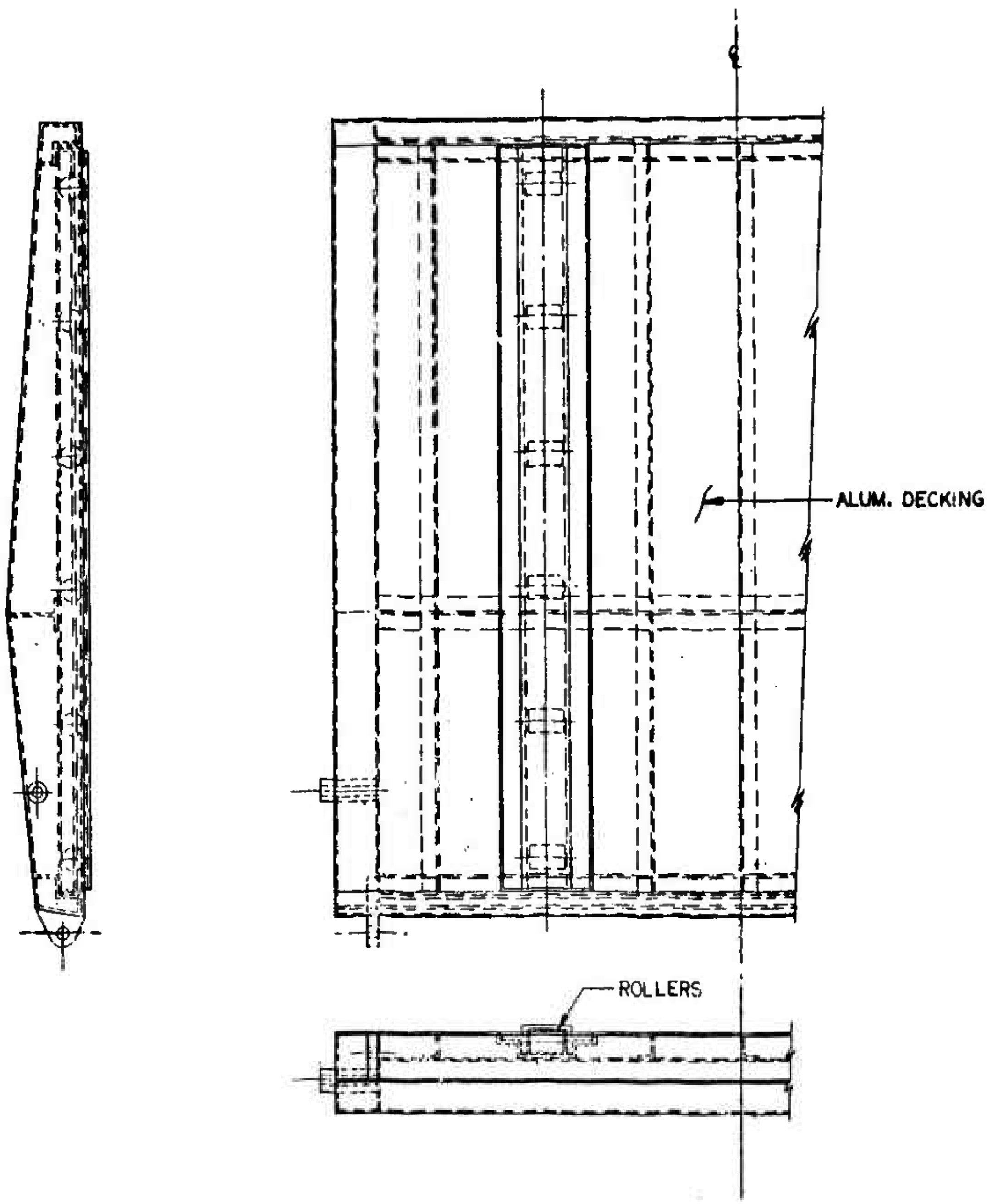


Figure 7-6 Ramp

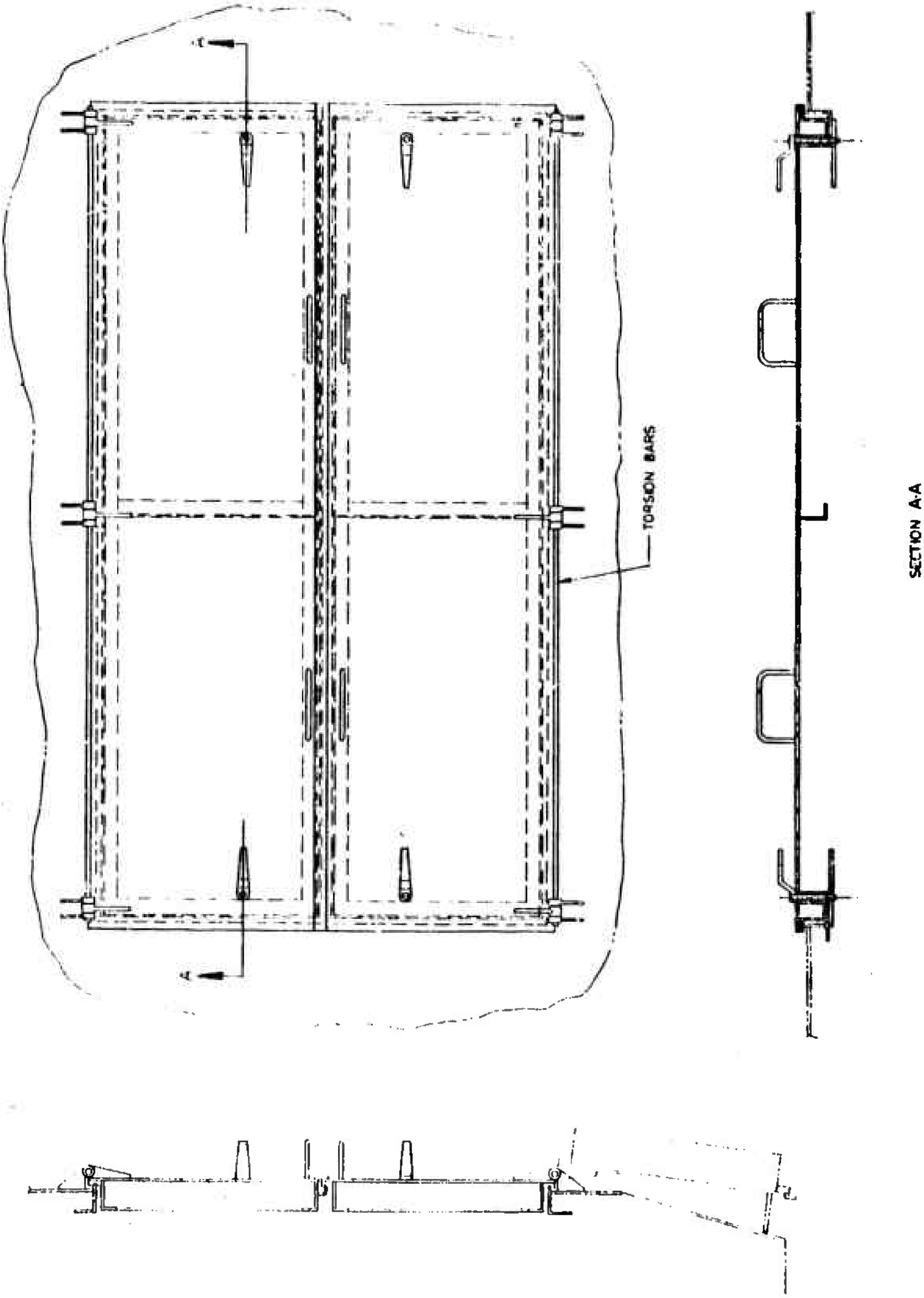


Figure 7-7 Cargo Hatch Doors

Bifold doors were considered but are not recommended because of the additional complexity in sealing and weight. With bifold doors two additional sealing surfaces are required. The bifold concept employs four doors, requiring more hinges and additional local framing, resulting in added weight and compexity.

7.4.3 Cargo Floor. The floor of the cargo compartment has approximately 84 sq. ft. of usable area for bulk and palletized cargo. Th facilitate cargo handling a conveyor system is built into it. This system consists of two longitudinal lines of rollers extending the full floor length. The rollers are mounted on a track, which in turn is recessed in the floor. The lateral spacing of the roller lines is compatible with the geometry of USMC standard pallets. When the cargo handling feature is unnecessary, such as when the vehicle is used as a personnel carrier, the roller track is inverted exposing a flat surface. Recessed tie downs in the floor are provided for securing cargo.

The floor is a welded assembly consisting of aluminum extrusions. The concept is shown in Figure 7-8. Three floor panels make up the total area. These bolt to the hull and can be removed independently for transmission, fuel tank, and bilge access.

7.4.4 Side Skirts and Fenders. Side skirts, bow fenders, and stern baffles are used on the track propelled concept while only side skirts are used for the auxiliary propelled concept. Their effect on water performance is described in Section 4.0. These items are non-ballistic and do not function as a part of the basic hull structure. The moveable parts of the bow fenders and stern baffles are fabricated from aluminum plate. They are actuated by hydraulic cylinders. The side skirts are aluminum and readily removable for suspension inspection, service, and repair.

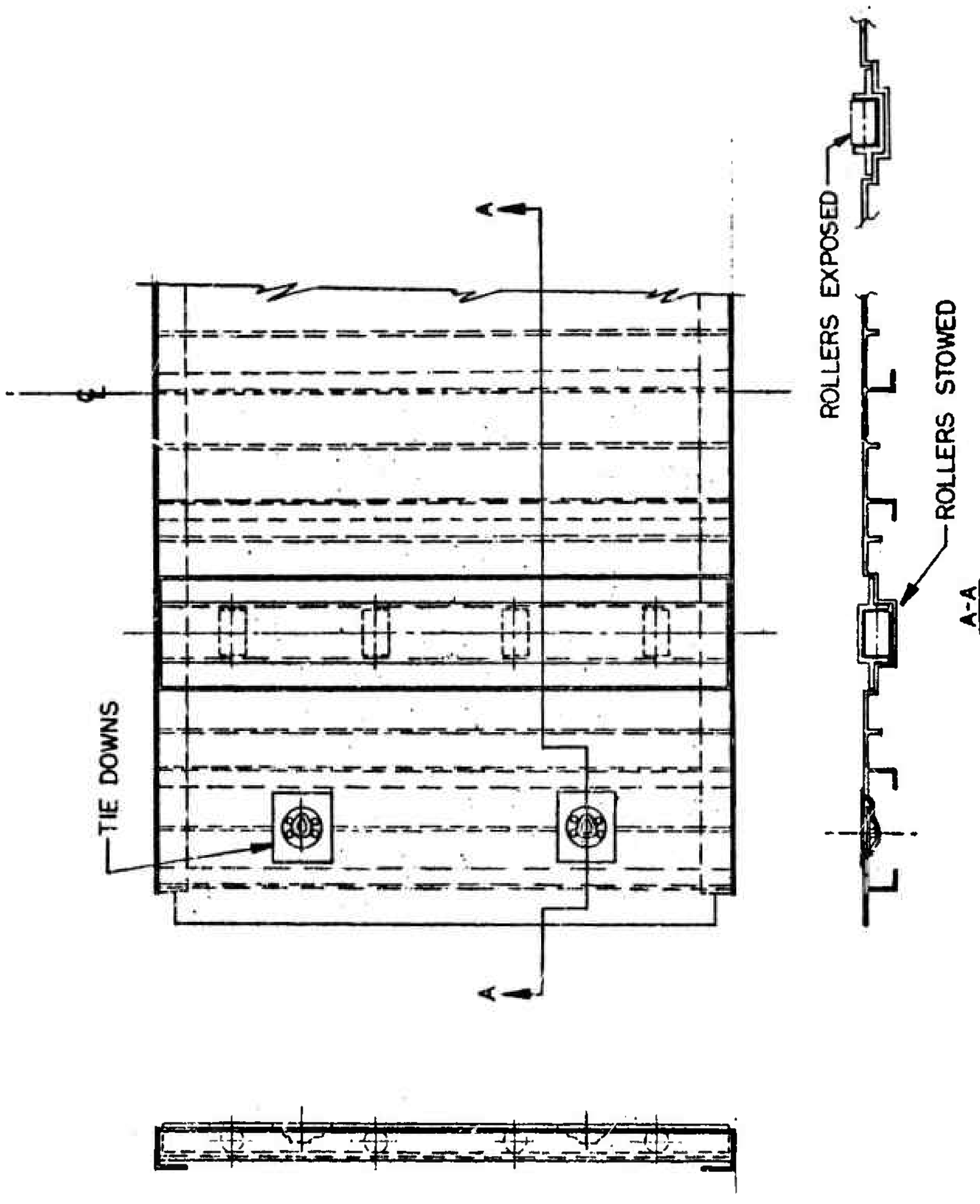


Figure 7-8 Cargo Floor

7.5 Hull Structure Weight. This section discusses the weight of hull, associated structure and the possible changes that may be expected for Phase II. The hull and structure represents approximately 39 percent of the GVW. This figure does not include the turret or OEM equipment. A further breakdown shows armor accounting for 61 percent of the hull structure weight and such items as paint, weld material, fasteners and standard hardware items collectively accounting for 9 percent. This leaves 30 percent of the hull and associated structure weight available for main frames, secondary frames, shrouding, floors, and miscellaneous supports.

The armor weight is based on steel; BHN 500 for sides, bow and stern and standard MIL-Spec armor for the top and bottom. The top armor is backed up by a nylon blanket. Section 6.0 provides the weight-cost-protection trade-off which led to this selection. The weight for paint and weld material is felt to be a realistic figure and is independent of material used for the basic hull. The remaining weight for framing and floors has been calculated based on the preliminary designs. The framing members and supports which are welded to the hull are steel. It is recognized that weight can be saved in framing when aluminum armor is used but, total evaluation of armor and framing indicated the weight saved in framing was offset by higher hull plating weight. This analysis is shown in Section 6.0. Several interior items such as floors and shrouding are independent of hull material and have been evaluated on the basis of optimum material for its intended function.

While final weight figures can only be known when the detailed design is completed, reasonable projections can be made by examining the variables involved and the degree to which they are known. Using the breakdown mentioned previously,

each category will be reviewed in light of possible variations. It is assumed that the vehicle geometry of basic shape, length, width, and height remains unchanged. Also, the tolerance on the various plates and shapes is not considered here.

The armor, which represents the greatest weight is governed by specification requirements. The present armor meets these requirements and is felt to be the best choice. Therefore, its weight is invariant unless ballistic protection levels are revised. The framing and support structure is subject to change in the final design stages due to interface clarification and better visibility in general. These changes for the most part are refinements to the initial design concept. It is assumed that these refinements add a tolerance to the preliminary weight figure of ± 10 percent. The miscellaneous items of paint, weld, etc. are also subject to further refinement but are more clearly known at the start. Therefore, a weight tolerance of ± 5 percent is assumed.

The result of these assumptions and effect on total GVW is as follows:

<u>Hull and Structure Weight Breakdown</u>

Armor	= 61 percent
Framing supports, floors, and shrouds	= 30 ± 3.0 percent
Paint, weld, fasteners, standard hardware	= 9 ± .45 percent

<u>Gross Vehicle Weight Percentages and Variance</u>

Armor	= 23.8 percent GVW
Framing supports, floors, and shrouds	= 11.7 ± 1.17 percent GVW
Paint, weld, fasteners, standard hardware	= 3.5 ± .18 percent GVW

Total hull and structure weight and variance potential	= 39 ± 1.35 percent GVW

7.6 Conclusions and Recommendations. The following conclusions and recommendations have been reached from the study of the hull and structure of the LVTPX12.

7.6.1 Conclusions.

1. The hull as a structural system should use semimonocoque construction.
2. The hull plating thicknesses which are defined by ballistic protection criteria are greater than required structurally for an optimized semimonocoque structure.
3. For all armor materials considered meeting specification requirement thicknesses, transverse frames are required.
4. The hull structural system consisting of armor plate and framing can be lighter in steel than in aluminum. Using the steel recommended is Section 6.0 versus 7039 aluminum, results in comparable hull weights but steel has a significant cost advantage.
5. The material used for interior items such as floors, baffles, shrouds and brackets can be different from the basic hull material. The material should be selected on the basis of satisfying intended function for least weight and cost. Electrolytic corrosion can be eliminated by painting, plating and applying sealer or paste on joining surfaces.
6. The structural loading criteria are based on vehicle operational requirements. The load conditions used will provide a given uniform level of structural integrity, but due to the statistical nature of the actual magnitude of the loads experienced, the true factors of safety are not known for any single vehicle.

7. The Marine Corps sling, drawing 5070, is not compatible with the LVTPX12 because the vehicle hoisting weight exceeds the rated sling capacity.

7.6.2 Recommendations.

1. To insure adequate design and a sound basis for weight evaluation, establish loading conditions, allowable stresses, and safety factors. This should be a joint venture between the user and contractor.
2. Additions and revisions to the preliminary specification for the LVTPX12 are as follows:
   (a) Define the methods of transportability and any applicable specifications.
   (b) Par. 3.12.1.6.2 Tie Down. Redefine. For example, specific tie downs required or can vehicle components such as suspension road wheel arms be used?
   (c) Par. 3.12.1.6.3 Lifting Devices. Redefine. The paragraph reads "capable of lifting the vehicle at combat weight." Combat weight is defined in paragraph 3.11 and is greater than the hoisting weight as defined in paragraph 3.11.9. Clarification is required. It is suggested that the use of Marine Corps sling, drawing 5070, be made a desirable requirement rather than mandatory.
   (d) Par. 3.11.9 Hoisting Characteristics. It is suggested that the first sentence be reworded to read: Hoisting weight of

the amphibian shall be the minimum practicable commensurate with the requirements of this specification.

(e) Paragraph 4.3.2 (c) Hoisting Pads and Slings. It is suggested that this paragraph include the vehicle condition for testing.

SECTION 8.0
POWER TRAIN

8.0 POWER TRAIN

The power train for an assault amphibian exerts a great influence on the design of the vehicle in terns of weight, space, and vehicle speed. The power train has the heaviest and largest units after the hull, and their location and selection has a vital influence on the vehicle. The selection of engine and transmission can proceed only when two primary questions are answered. These are:

- What do we need?
- What is available?

For the water speeds being considered, the first question was largely answered in Section 4.0 since the power required on water exceeds that required on land by a wide margin. This section is a survey of all available power systems and selection of systems for the LVTPX12. The following pertinent subjects were examined in the course of this study.

- A survey of available engines
- Comparison and evaluation of engines
- A review of available drive systems
- Selection of drive systems
- Design of the power package for a track-propelled LVTPX12
- Design of the power package for an auxiliary-propelled LVTPX12

Definition of terms are in Appendix D.

8.1 Power Train Requirements. In addition to the power package requirements as prescribed in the BUREAU OF SHIPS PRELIMINARY SPECIFICATION FOR ASSAULT AMPHIBIAN PERSONNEL CARRIER (LVTPX12), dated 31 January 1964, there are implied requirements based on conditions that the vehicle must satisfy. For a 50,000 pound vehicle, an engine of at least 300 HP is needed to drive the vehicle at 30 MPH on the level or at 2 MPH on a 60 percent slope. The water speed requirements are more stringent. It would take an 800 HP engine to drive a 50,000 pound track-propelled vehicle at 8 MPH in the water, and almost as much power for a propeller-driven vehicle at 10 MPH. All new tracked-vehicle designs provide for a tractive effort at stall equal to the vehicle weight, and the LVTPX12 should be no exception. The land speeds quoted earlier are the minimum in the specification. The trend in tracked vehicles is to make them more agile; therefore, another implied requirement is to exceed the minimum speeds of 30 MPH on the level, and 2 MPH on the slope.

Additional implied requirements for the transmission besides the items covered in the preceding paragraph include braking and steering. The vehicle brakes must hold on the 60 percent slope and provide a deceleration rate on the level of at least 16 ft/sec./sec. The vehicle must be able to make a 90 degree turn on a 60 percent slope at 2.5 MPH.

The designs of newly developed tracked vehicles provide for the ability to apply 70 percent of the total available power to one track. In order to have good steering and control in the water, it is essential that the differential track speed be at least 10 MPH, and preferably, 20 MPH. The lubrication

systems of both the engine and transmission must operate satisfactorily on 60 percent slopes, fore, aft and side.

8.2 Review of Available Engines. Among the possible power sources that could be used are fuel cells, steam engines, and nuclear power units. However, these are not sufficiently developed to provide assurance that prototype units could be available in the program time frame, and therefore, they are dropped from further consideration. Conventional spark-ignition engines such as used in the LVTP5 are also dropped primarily because of their dependence on a single type of fuel and fuel economy poorer than the diese engine. Compounding of diesel engines and gas turbines offer advantages, but the lack of available prototypes within the time frame eliminates them from consideration.

The available gas turbines are very attractive due to high power for their weight and size. The major drawbacks are their high air consumption (several times that of a diesel engine), high noise level for intake and exhaust, high fuel consumption (50 to 100 percent more fuel consumed than a diesel engine at full load), and vulnerability to salt deposits in the compressor which cause severe power drops. In surfing, it is possible for the vehicle to be under water for as long as 18 seconds, and the high air consumption of a gas turbine makes the air supply problem several times greater than that of the diesel engine. Silencing problems of turbines will be critical due to the proximity of personnel. For the preceding reasons, the simple cycle gas turbines are excluded from further consideration.

The regenerative gas turbines now being developed solve the fuel consumption problem, and decrease the exhaust noise problem, but cannot be available

soon enough to meet the build schedule of the LVTPX12. Nevertheless, these engines will be strong contenders a few years from now. They should be reconsidered for advanced versions of the LVTPX12.

The engines most suitable for the LVTPX12 are conventional diesel (or compression-ignition) engines. A survey of what is available follows.

8.2.1 Continental. Continental Motors Corporation has a license from the British Internal Combustion Engine Research Association, better known as BICERA, to use the Variable Compression Ratio (VCR) piston in their engines. The piston automatically changes the compression ratio over a 2 to 1 spread according to engine loads. VCR provides added output potential up to three times as high as that which is considered high today. Multi-cylinder engines with power levels in excess of 1 HP/cu.in. are already under test. Continental estimates that by 1972, engines will be available with outputs of 1.8 HP/cu.in. displacement.

The high power is obtained by turbocharging and aftercooling. The VCR principle is of very little advantage to a naturally aspirated diesel engine, because the power lost by the compression ratio change could not be recovered except by supercharging. The fuel economy of a VCR engine is somewhat less than that of a fixed compression ratio engine at full load, but is slightly better under light load. The engine always starts at maximum compression ratio, which improves starting performance. The VCR engine at present (1965) is being developed to use diesel and CITE fuels only, but the engine could be further developed to burn gasoline. One of the major problems that exists

in the combustion of gasoline is light load operation. A high compression ratio is required to insure consistent ignition of gasoline, and VCR automatically provides it.

At this time, the AVDS-1100 VCR engine appears to be the leading contender for powering the US/FRG Main Battle Tank. A summary of available and concepted engines from this supplier are listed in Figure 8-1. All the engines are U.S. Government owned designs and can be procured competitively.

8.2.2 Detroit Diesel. The Detroit Diesel Division of the General Motors Corporation has four cylinder sizes and can furnish engines with almost any number of cylinders and a wide range of horsepower. All of these engines use the two stroke cycle principle. The higher horsepowers are obtained by turbocharging. The three smaller sizes in production are the 53-cubic inch, 71-cubic inch, and 110-cubic inch engines. The 149-cubic inch cylinder is being developed for a V-8 engine. Detroit Diesel can supply the 53 and 71 series engines in multifuel versions. At present, they do not have a fuel density compensator on the fuel injection system so that the engine power output will vary as the specific weight of the fuel. In commercial applications, ether capsules are used to start the engines at low temperatures. The U.S. Army has specified that use of engine starting fluids are not allowed, and large fleet operators in cold climates have found through experience that starting fluids lead to engine damage. For these reasons, Detroit Diesel has adopted a flame heater for their military service engines. Reliable starting with these heaters below -25 degrees Fahrenheit has not been definitely proved.

MODEL	TYPE	COOLING	GHP	RPM	WEIGHT	L	W	HT	PRESENT** STATUS	PLANNED USAGE	PROTO. AVAIL.
AVDS-750VCR	C.I.	AIR	750	2800	2553	43	39	38	DEV		14 MO.
AVDS-1100VCR	C.I.	AIR	1120	2800	3080	58	39	34	DEV	MBT	6 MO.
AVDS-1195VCR	C.I.	AIR	1195	2400	3931	54	44	41	PROP		25 MO.
AVDS-1490VCR	C.I.	AIR	1490	2400	4467	58	44	41	PROP		25 MO.
LDS-465	C.I.	LIQUID	210	2800	1561	48	24*	40	XM656		IN PROD.
LDS-465VCR	C.I.	LIQUID	450	2800	1692	48	24*	40	PROP		14 MO.
LVDS-1790	C.I.	LIQUID	750	2400	4517	67	44	41	PROP		25 MO.
AVDS-1790-2A	C.I.	AIR	750	2400	4429	67	44	41	LVT-E1, M60A1, ETC.	LVTP5	IN PROD.
AVDS-1790VCR	C.I.	AIR	1790	2400	4759	67	44	41	PROP		20 MO.
AVDS-370VCR	C.I.	AIR	350	2800	2097	25	39	33	PROP		25 MO.
AVDS-550VCR	C.I.	AIR	550	2800	2317	32	39	33	PROP		25 MO.
AVDS-930VCR	C.I.	AIR	930	2800	2808	50	39	34	PROP		25 MO.

* STD. CONFIGURATION 29 INCHES WIDE

** DEV - IN DEVELOPMENT

PROP - PROPOSED

TYPE C.I. - COMPRESSION IGNITION

Figure 8-1 Available Continental Engines of U.S. Government Owned Designs

The Detroit Diesel engines are desirable candidates because the military operates large numbers of Series 53 and 71 engines. These engines are the most compact, and lightest liquid-cooled engines available. Since most versions are in production there are no development costs, the prototype costs are low, and the production costs are reasonable. The designs are proprietary with General Motors.

A summary of the engines considered is shown in Figure 8-2. The 110-cubic inch size has been deleted because this is an older design and is only available in limited combinations.

8.2.3 Caterpillar. The Caterpillar Tractor Company is developing a family of engines using the VHO (Very High Output) concept. The development is being directed and funded by ATAC. The goal is to produce engines of four, six, eight, and twelve cylinder configurations that will develop 1 HP per cubic inch. All engine designs provide for the use of a common cylinder. The engines use the four stroke cycle principle and obtain the high power with pre-combustion chambers, fixed compression ratio, turbocharging, and after-cooling. These engines are light, compact, and very narrow. Power can be taken off either end of the engine, and accessory drive locations are available at each corner of the engine. The program is funded to build and develop the 12 cylinder version. The design layouts for the other three members of the family are revised as the 12 cylinder engine design changes. The first 12 cylinder engine was produced in November, 1964, and is used in laboratory development work.

MODEL	TYPE	COOLING	GHP	RPM	DRY WEIGHT	L	W	HT	PRESENT STATUS	PLANNED USAGE	PROTC. AVAIL.
12V71T	C.I.	LIQUID	800	2500	2512	54	40	42	COM**		6 MO.
16V53T	C.I.	LIQUID	800	2800	3240	65	37	38	COM		12 MO.
8V149T	C.I.	LIQUID	1200	2400	6000	47	44	42	COM		18 MO.
8V53T	C.I.	LIQUID	400	2800	1258***	37	37	38	COM		4 MO.
12V53T	C.I.	LIQUID	600	2800	2400	49	37	38	COM		8 MO.
6-71T	C.I.	LIQUID	400	2500	1802	53	25	48 ****	BARC BOATS, ETC.		4 MO.
8V71T	C.I.	LIQUID	530	2500	1903	45	40	42	M107, M110, M108, ETC.	AIFV	12 MO.
6V53T	C.I.	LIQUID	300	2800	1237	32	37	39	XM551, LARC-V		4 MO.
4-53T	C.I.	LIQUID	200	2800	1410	35	23	33			12 MO.
3-53T	C.I.	LIQUID	150	2800	870	28	23	33	*		8 MO.

* THE NATURALLY ASPIRATED VERSION IS USED IN THE XM561
** COMMERCIAL
*** CAST IRON BLOCK
****ESTIMATED MINIMUM HEIGHT FOR WET SUMP IS 41 INCHES
ESTIMATED MINIMUM HEIGHT FOR DRY SUMP IS 35 INCHES

Figure 8-2 Detroit Diesel Engines Considered for LVTPX12

The design, except for the fuel system, is Government-owned. ATAC is planning to have other fuel injection equipment available for the engines to make them completely Government-owned designs. The engines are multi-fuel. The Caterpillar engineers have rated their engines for LVTPX12 use at only 60 horsepower per cylinder, whereas for land vehicle use the power level is 80 horsepower per cylinder. This derating makes the VHO engines less attractive for LVTPX12 usage. General characteristics of the four VHO engines and two earlier engines which have been developed for ATAC are tabulated in Figure 8-3.

8.2.4 Cummins. The Cummins Engine Company is in an enviable position in the commercial truck field in that it is the supplier of over half of all diesel engines used in over-the-road trucks. The engines listed in the tabulation in Figure 8-4 are the newest in the Cummins engine line. The engines are relatively simple and efficient. Cummins has no plans to produce the engines in aluminum, at present, but if they do, the engines will be among the lightest in their type class. The VOOM and VOOMER engines are being redesigned. The values shown in Figure 8-4 are preliminary estimates. The engines are designed to operate on CITE and diesel fuels. Jet fuels can be used as alternates, if the CITE specifications are met. Gasoline can be used as an emergency fuel only. The engine designs are proprietary with Cummins.

8.2.5 Curtiss-Wright. The Curtiss-Wright Corporation has a large, lightweight diesel engine that was originally designed and built by the Packard Motor Car Company for Navy minesweeper use. This engine design is owned by the Government and the engine can be multi-sourced. In addition to this,

MODEL	TYPE	COOLING	GHP	RPM	DRY WEIGHT	L	W	HT	PRESENT* STATUS	PLANNED USAGE	PROTO. AVAIL.
LMS-350VHO	C.I.	LIQUID	240	2800	1472	36	23	36	DES		24 MO.
LMS-525VHO	C.I.	LIQUID	360	2800	1922	48	23	36	DES	MICV	24 MO.
LVMS-700VHO	C.I.	LIQUID	480	2800	2232	39	32	36	DES		24 MO.
LVMS-1050VHO	C.I.	LIQUID	720	2800	2885	51	32	36	DEV		9 MO.
LVDS-1100	C.I.	LIQUID	700	2200	2697	52	48	44	DEV		12 MO.
LDS-750	C.I.	LIQUID	470	2200	1797	46	35	43	DEV		15 MO.

*DES - DESIGN

DEV - DEVELOPMENT

Figure 8-3 General Characteristics of Engines Developed for U.S. Government by Caterpillar

MODEL	TYPE	COOLING	GHP	RPM	DRY WEIGHT	L	W	HT	PRESENT STATUS	PLANNED USAGE	PROTO. AVAIL.
VT6-300	C.I.	LIQUID	350	2600	1740	35	32	39	COM.*	COM., UET	IN PROD.
VT8-400	C.I.	LIQUID	400	2600	2060	43	32	39	COM.,UET	UET	IN PROD.
VIMMER T	C.I.	LIQUID	600	3100	3100	66	32	39	NONE	COM.	9 MO.
VOOM T	C.I.	LIQUID	1000	2300	4600	66	41	43	NONE	COM.	18 MO.
VOOMER T	C.I.	LIQUID	1500	2300	6200	89	41	43	NONE	COM.	18 MO.

*COMMERCIAL

Figure 8-4 Cummins Engines Characteristics

Curtiss-Wright has the North American license for the NSU-Wankel Rotating Combustion (RC) Engine. In 1958, Curtiss-Wright started design of an experimental Wankel engine eight times as large as the NUS-Wankel demonstration engine. The RC1-60 was operated for the first time in March 1959. Since then, RC engines have accumulated over 20,000 test hours. Testing since 1961 has been without supplementary oil mixed in the fuel. Engines have been built and tested in the displacement range of 4.3 to 1920 cubic inches. Over 150 HP and 160 bmep have been demonstrated with the basic RC1-60 test rig engine. In addition, working with the RC1-60, they have established the feasibility of high output, air cooling, and multi-fuel capability.

The RC engine's strongest claim is large output from a small, and consequently lightweight package. The engine operates on the 4-stroke cycle, even though it is ported, and each rotor bank delivers one power stroke per revolution. The combination of small size and simple components gives an apparent cost advantage in production. (There has been no production or tooling set up yet for production.) The RC2-60 engines have been evaluated in automobiles by the automobile manufacturers, in small boats, in Air Force ground power units, and other applications. ATAC is currently (1965) negotiating with Curtiss-Wright on the installation of an engine in an M35, 2-1/2 ton truck. Curtiss-Wright has started the design of an air-cooled 4 rotor aircraft engine, model RC4-90. They expect to receive a contract from the Navy, to be jointly funded by the Marine Corps and Navy, in the spring of 1965. This engine will have fuel injection and will have the capability of burning gasoline, jet fuels, and light oils.

The greatest drawback to the use of the RC engines in the LVTPX12 is the fact that development is required which is expensive in terms of both money and time. In addition, the designs are proprietary and cannot be multi-sourced. There is some question as to whether prototype engines could be available soon enough for the prototype LVTPX12. An air-cooled RC4-90 would require very little development, but a liquid-cooled version would require development. If the aircraft engine development proceeds according to the present schedule, there will be a very good chance that engines could be available for the LVTPX12. Curtiss-Wright engines' characteristics are presented in Figure 8-5.

8.2.6 <u>Lycoming</u>. Lycoming Division of Avco Corporation, with ATAC funding and direction, is developing a family of three multi-fuel, compression-ignition, two stroke, valveless engines of four, six, and eight cylinders. The preliminary development of the small engine is complete and the engine will be installed in new military vehicles, such as the XM561, 1-1/4 ton truck, within the next several months (1965). The eight-cylinder engine, slightly behind in development, is planned for larger vehicles such as the XM656, five ton truck. The six-cylinder engine development is the least advanced of the three. These engines are very light weight for their power output. The efficiency of the engines will lie somewhere between that of spark-ignition gasoline engines and that of conventional compression-ignition engines, because the design was directed toward the goals of light weight and simplicity. These engines are of Government owned designs and can be multi-sourced.

Specifications for these engines are shown in Figure 8-6.

MODEL	TYPE	COOLING	GHP	RPM	DRY WEIGHT	L	W	HT	PRESENT*** STATUS	FUTURE USAGE	PROTO. AVAIL.
12V-142	C.I.	LIQUID	750	2300	4050	79	43	53	NAVY MINE SWEEPER		9 MO.
RC4-60	ROT. COMB	LIQUID	270	5000	550	35	24	22	PROP		18 MO.
RC6-308	ROT. COMB	LIQUID	1200	3070	1882	66	30	31	PROP		24 MO.
RC2-60	ROT. COMB	LIQUID	180	5000	270	18	22	22	DEV	COM	5 MO.*
RC4-90	ROT. COMB	LIQUID	405	5000	702	41	24	22	PROP		9 MO.**
RC4-90	ROT. COMB	AIR	405	5000	466	51	23	23	DES	COIN AIRCRAFT	8 MO.**

* CARBURETED VERSION, FUEL INJECTION VERSION IF COIN PROGRAM FUNDED.

** IF THE AIR-COOLED RC4-90 IS FUNDED FOR COIN AIRCRAFT.

*** PROP - PROPCSED

DEV - DEVELOPMENT

DES - DESIGN

Figure 8-5 Curtiss-Wright Engines Characteristics

MODEL	TYPE	COOLING	GHP	RPM	WEIGHT	PRESENT STATUS*	FUTURE USAGE	PROTO. AVAIL.
AVM-410	C.I.	AIR	160	2600	667	DEV	XM561	6 MO.
[illegible]	C.I.	AIR	250	2600	857	PROP	XM410	18 MO.
AVM-625	C.I.	AIR	325	2600	1067	DEV	XM656	8 MO.

* DEV - DEVELOPMENT

PROP - PROPOSED

Figure 8-6 Lycoming Engines' Characteristics

8.2.7 Harnischfeger. Harnischfeger engine, designed and originally built by the P. and H. Corporation, was transferred to NAPCO Industries, Inc. and is now marketed by the latter corporation's Diesel Division. The engine is a diesel of conventional design featuring light weight construction. The engine was proposed to the Navy for use in minesweepers. This engine is proprietary and cannot be multi-sourced. Pertinent characteristics are listed in Figure 8-7.

MODEL	TYPE	COOLING	GHP	RPM	DRY WEIGHT	PRESENT STATUS	FUTURE USAGE	PROTO. AVAIL.
V12	C.I.	LIQUID	600	2050	3900	DEV*	MINE SWEEPER	6 MO.

* DEV - DEVELOPMENT

Figure 8-7 NAPCO Engine Characteristics

8.2.8 Fairbanks Morse. Fairbanks Morse, Incorporated distribute, in the United States, a large diesel engine of English design. The engine has a V-12 arrangement, is of four stroke cycle, and has turbochargers and aftercoolers. The engine is in production and available but is of proprietary design and cannot be multi-sourced. The specifications are outlined in Figure 8-8.

MODEL	TYPE	COOLING	GHP	RPM	DRY WEIGHT	PROTO. AVAIL.
120TCW	C.I.	LIQUID	910	1800	7600	6 MO.

Figure 8-8 Fairbanks-Morse Engine Characteristics

8.2.9 Hercules, Waukesha, Mack, and Allis-Chalmers. Hercules Engine Division of the Hupp Corporation, Waukesha Motor Co., Mack Trucks, Inc., and Allis-Chalmers Manufacturing Co. have diesel engines (compression-ignition) that are used extensively in the commercial field. All of these engines are designed to commercial standards of long life, low fuel consumption, low cost, and liquid cooling. A common disadvantage is that they are all large and heavy. These engines are all of proprietary design and cannot be multi-sourced. Selected engine data from the companies are listed in Figure 8-9.

MFR.	MODEL	GHP	RPM	L	W	H	DRY WEIGHT	PROTO. AVAIL.
HERCULES	D-426T	180	2600	44	24	36	1597	3 MO.
HERCULES	D-298HT	140	2800	38	20	32	970	3 MO.
WAUKESHA	135-DKBS	185	2800	46	25	34	1577	4 MO.
WAUKESHA	148-DKBS	280	2100	55	25	47	2577	4 MO.
WAUKESHA	F1197DSI	485	1800	65	31	51	4092	4 MO.
MACK	ENDT864	350	2300	45	37	44	2377	3 MO.
A-C	3500	230	2400	44	23*	32	1457	3 MO.
A-C	25000	435	2100	58	30	53	3457	3 MO.

* 20 Inches Available and 19 Inches possible.

Figure 8-9 Other Commercial Engines Charecteristics

Weight and power compilations for the United States engines are in Appendix D.

8.2.10 <u>European Manufacturers</u>. Many European engine manufacturers have compression-ignition engines of types that should be considered for the LVTPX12. These are units that are in production, readily available, and several makes are used in military vehicles. Many are distributed in the United States. Data on a selected group are listed in Figure 8-10.

8.3 <u>Initial Trade-off - Power Plants</u>. The preceding paragraphs have described a large number of engines suitable for the LVTPX12. The following guidelines for selection of an engine were considered:

- Weight
- Size
- Human factors (noise, vibration, radiated heat, fumes)
- Ease of maintenance
- Availability of prototypes
- Complexity of installation
- Necessary power
- Sensitivity to operational environment (salt, humidity, temperature)
- Cost
- Simplicity of parts inventory
- Reliability
- Safety
- Quantity, cost, and availability of fuel
- Compatibility with the existing and the continuing military supply system.
- Multifuel capability

MFR.	MODEL	COOLING	GHP	RPM	DRY WEIGHT	USAGE
1. DEUTZ	8F6M716	LIQUID	230	1800	2403	COM.*
DEUTZ	8F12M716	LIQUID	460	1800	4189	COM.
DEUTZ	F8L714	AIR	195	2300	2039	COM.
DEUTZ	F12L714	AIR	290	2300	2914	COM.
2. HISP.	HS110	LIQUID	710	2600	3000	FRENCH TANK AMX30
3. LEY.	L60	LIQUID	700	2100	4250	BRITISH CHIEFTAIN TANK
4. M.A.N.	D1548M7	LIQUID	190	2000	2180	COM.
M.A.N.	D1546M5	LIQUID	170	2200	1540	COM.
5. MER.	MB833	LIQUID	820	2200	3638	COM.
MER.	MB837	LIQUID	600	2200	2977	COM.
MER.	[illegible]	LIQUID	450	2200	2514	COM.
MER.	OM346	LIQUID	220	2200	1500(E)	COM.
6. PER.	6.354	LIQUID	130	2800	836	COM.
7. R.R.	K60	LIQUID	240	3750	1560	COM. & SWEDISH TANK STRIDSVAGN S

* COMMERCIAL

1. KLOCKNER-HUMBOLDT-DEUTZ AG, WEST GERMANY
2. HISPANO-SUIZA, FRANCE
3. LEYLAND MOTORS LIMITED, ENGLAND
4. MASCHINENFABRIK AUGSBURG NURNBERG AG, WEST GERMANY
5. DAIMLER-BENZ AG, WEST GERMANY
6. PERKINS GROUP OF COMPANIES, ENGLAND
7. ROLLS-ROYCE, ENGLAND

Figure 8-10 European Engines' Characteristics

- Compression Ignition
- Interchangeable with other vehicles
- Multi-Source capability

The engine to be selected for the US/FRG Main Battle Tank has a great influence on the engine to be used in the LVTPX12. The two programs mesh well in time and the use of a common engine would be of great advantage to the military system. At the time this is written (April 1965) there has been no decision released as to what engine will be used in the MBT. Originally there were three strong U.S. contenders: the Detroit Diesel 12V71T, the ATAC funded Continental AVDS-1100VCR, and the ATAC funded Caterpillar LVMS-1050VHO. The original power level was 750 horsepower but it has since increased to something over 1200 horsepower. It is probable that the U.S. engine choice has been narrowed down to a version of the AVDS-1100VCR. This engine is built and is presently under test. Continental is working to increase the power to 1400 horsepower. In addition, an air cooled engine has many advantages over a liquid cooled engine for the MBT. There is a very strong likelihood that the AVDS-1100VCR engine will be the MBT engine and that the power level will be near 1400 horsepower.

Because the leading contender for the US/FRG MBT appears to be an air cooled engine, it would be of value to outline some of the good and bad points of liquid and air cooling. Air cooling advantages are:

- Elimination of liquid cooling media: pumps, radiator, surge tanks, piping, and associated connections.
- Simplification of logistics: no fresh water, antifreeze, or rust inhibitors.

- Cooling system maintenance is minimized.
- Engines are compact and light weight when the power package with cooling system is considered.

Air cooling disadvantages are:

- Necessity of single cylinder construction, finning, and baffles causing cylinder spacing to be larger than comparable liquid cooled engines and, thus, increasing engine size.
- Requirement for large power absorbing cooling fans.
- The fins and oil coolers are vulnerable to dirt deposits. Often the oil coolers are mounted on the engine and are susceptible to oil spills or leaks which collect dirt.
- The aluminum fins operate at a high temperature and can be corroded by salt water.
- For amphibian use either salt water has to be removed from the air, or a sea water cooler is needed so that the air can be recirculated. Under these conditions the engine fans and sea water pumps are operating and require power.
- Air cooled engines often have a greater heat rejection to the lubricating oil.
- Currently developed air cooled diesels have the problem of retaining enough engine heat to keep the engine idling at low temperatures.

Liquid cooling advantages are:

- Engines are small, light, and compact.
- Liquid cooling provides an efficient cooling medium for an engine.

- Engine heat can be retained in the engine and controlled during low temperature idling.
- Radiators can be put wherever convenient. In an amphibian the radiators can be in the water providing maximum cooling when most needed, along with the opportunity of decreasing fan power requirements.
- The engine gross power represents the power available <u>with</u> the water pump operating.

Liquid cooling disadvantages are:

- Logistics: water, antifreeze, and rust inhibitors must be supplied.
- More cooling system problems exist due to additional components and leaks.
- System is vulnerable to air entrainment requiring more attention during design and service.
- For cold weather operation some form of shutter shield may still be needed to retain the heat.

In order to make a quantitative evaluation and selection of the remaining engines, either diesel or rotating combustion engines, guidelines must be established.

The following are the guidelines for single engine - track propulsion.

- The lowest power level (GHP) that can be considered is 759 HP for a liquid cooled engine or 855 HP for an air cooled engine. (Refer to calculations in Appendix D.)
- On a weight comparison based on pounds per horsepower, all engines over four pounds per horsepower are eliminated. (This is based on state-of-the art developments in engines, See Figure 8-11.)

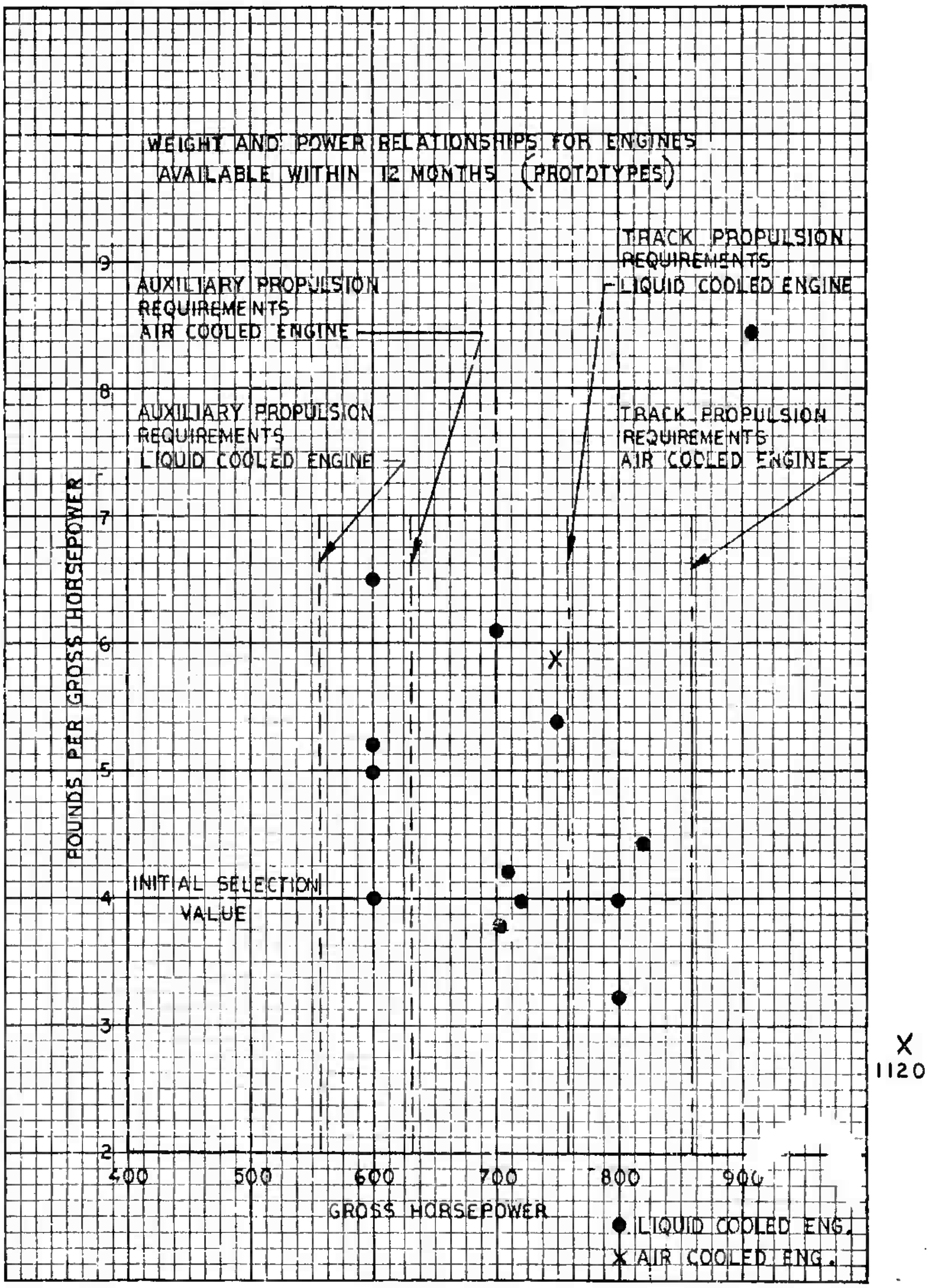


Figure 8-11 Specific Weight - Engines

- In order to meet the prototype build schedule, any engine considered must be available in nine months or less.

Figure 8-12 shows the engines remaining after application of these criteria.

MFR	MODEL	GHP	DRY WT. #/HP	PROTO. AVAIL.	MULTI-SOURCE
CONTINENTAL	AVOS-1100VCR	1120	2.8	6 MO.	YES
DETROIT DIESEL	12V71T	800	3.2	6 MO.	NO

Figure 8-12 Initial Selection for Single Engine - Track Propulsion

The following are the guidelines for single engine - auxiliary propulsion.

- The lowest power level (GHP) that can be considered for the propeller-driven craft at 10 MPH is 580 HP for a liquid-cooled engine or 655 HP for an air-cooled engine. (Refer to calculations in Appendix D.)

- On a weight comparison based on pounds per horsepower, all engines over four pounds per horsepower will be eliminated. (This is based on state-of-the- art development in engines.)

- Any engine considered must be ready for delivery in nine months or less in order to meet the prototype build schedule.

Figure 8-13 shows the engines remaining after application of these criteria.

MFR	MODEL	GHP	DRY WT. #/HP	PROTO. AVAIL.	MULTI-SOURCE
CONTINENTAL	AVOS-1100VCR	1120	2.8	6 MO.	YES
DETROIT DIESEL	12V71T	800	3.2	6 MO.	NO
CATERPILLAR	LVMS-1050VHO	720	4.0	9 MO.	YES

Figure 8-13 Initial Selection for Single Engine - Auxiliary Propulsion

The following are the guidelines for dual engine installation - track propelled.

- The smallest power level (GHP) that could be considered is 380 HP (liquid-cooled engine) based on two engines supplying a total of 760 horsepower or 428 HP (air-cooled engine), based on two engines supplying a total of 856 HP.
- All engines that weigh more than 4.5 pounds per horsepower are eliminated, based on state-of-the-art engine developments in these smaller size engines.
- The engine must be ready for delivery in nine months from the date of order to meet the prototype build schedule.
- In order to fit over the tracks the engine must not be more than 25 inches wide.

The engines remaining after application of these factors are shown in Figure 8-14.

MFR.	MODEL	GHP	DRY WT. #/HP	WIDTH	PROTO. AVAIL.	MULTI-SOURCE
DETROIT DIESEL	6-71T	400	4.5	25	4 MO.	NO
CURTISS-WRIGHT	RC4-60(LIQ.)	405	1.7	24	9MO.*	NO

* AVAILABILITY IF AIR-COOLED ENGINE IS DEVELOPED FOR COIN AIRCRAFT STARTING IN LATE SPRING 1965.

Figure 8-14 Initial Selection for Dual Engine Installation - Track Propelled

The following are the guidelines for dual engine installation - auxiliary propulsion.

- The smallest power level (GHP) that could be considered is 290 HP for a liquid cooled engine (two engines supplying 580 HP) or 328 HP for an air cooled engine (two engines supplying 655 HP).
- All engines weighing more than 4.5 pounds per horsepower are eliminated.
- The engine must be available for delivery in nine months or less.
- The maximum engine width is 25 inches.

The following engines listed in Figure 8-15 meet these standards.

MFR.	MODEL	GHP	DRY WT. #/HP	WIDTH	PROTO. AVAIL.	MULTI-SOURCE
DETROIT DIESEL	6-71T	400	4.5	25	4 MO.	NO
CURTISS-WRIGHT	RC4-90(LIQ)	405	1.7	24	9 MO.*	NO
CURTISS-WRIGHT	RC4-90(AIR)	405	1.2	23	4 MO.*	NO

* AVAILABILITY IF ENGINE IS DEVELOPED FOR COIN AIRCRAFT STARTING IN LATE SPRING 1965.

Figure 8-15 Initial Selection for Dual Engine - Auxiliary Propulsion

8.4 Review of Available Drive Systems. The three general types of systems that could be used for the LVTPX12 are: mechanical, hydrostatic, and electric drive. The review of mechanical drive transmissions will be restricted to power-shifted transmission that have steering function and use a torque converter to engage the load. The hydrostatic drive review will be restricted to those hydrostatic transmissions available. Electric drive has interesting possibilities, but at present, it appears the units would be too large and heavy for the LVTPX12. A detailed study of electric drive which justifies this conclusion is included in Appendix D.

For tracked vehicles that operate at speeds more than a few miles per hour, some form of regenerative steering is required. When the top speed exceeds 30 MPH, a smooth, continuous,regenerative steer is required. The new power-shifted transmissions for high-speed tracked vehicles incorporate hydrostatic steering. These systems allow the vehicle to turn on a radius rather than by straight-line segments. Hydrostatic transmissions incorporate hydrostatic steering, and electric drive accomplishes the same function electrically.

8.4.1 Hydro-Mechanical Transmissions.

8.4.1.1 Allison. Allison Division of the General Motors Corporation has been the prime source of power-shifted tracked-vehicle transmissions for the military for the past 20 years. Their transmissions feature regenerative steering, planetary gear sets, torque converters, and large hydraulically operated clutches.

Of the transmissions currently available, none were designed for use in an amphibian. The Allison CD-850 was originally designed for use in the M46 and M47 tanks and is currently being used in the M60 tanks. It was adopted for use in the LVTP5 because it was developed, tooled, and in production. It is the only transmission (in various versions) in use in production military track laying vehicles of the weight class from the LVTP5, or 87,000 pounds and upward. This transmission has two speeds forward and one speed in reverse. A torque converter (hydrodynamic) of the poly-phase type with a high torque ratio at stall (approximately 3.6 to 1) and no-lockout clutch is used. Geared steering is accomplished through a triple differential. The steering function is suitable for a moderate-speed track laying vehicle(up to 30 MPH) on land, however, the water steer requirement is not met satisfactorily.

Another transmission currently in production is the Allison XTG-411. This is a four speed steering transmission which incorporates a torque converter with lockout clutch. Geared steering is utilized in the top two gears and clutch brake steering is used in first and second gear.

The XT1400 series transmission has been used in heavy vehicles and is currently out of production. It has three speeds forward and one in reverse, and has double differential steering. This transmission is the heaviest of the group.

At a lower power level the XTG-250 steering transmission is also a current design, although not yet in production. This steering transmission has four speeds forward and two in reverse, and incorporates a torque converter and lockout clutch. Geared and pivot steering is incorporated.

The GS400 steering unit has been used in the Universal Engineer Tractor. This unit has two ranges forward only, one for water operation and the other for land operation. Geared steer is used in the land range and clutch-brake in the water range. The unit is quite heavy for the small power capacity.

These five units are the only transmissions that are currently available.

Allison is developing the X-700 Transmission which is a four-speed forward, four-speed reverse with lockup torque converter and hydrostatic steering. Two smaller versions of this transmission X-300 and X-500 are being tested (X-300) and designed (X-500). The Allison transmissions, possible for use in the LVTPX12, are shown in Figure 8-16. More details on the transmissions are in Appendix D.

	HP		DRY		PROTO.	MULTI-
	WATER	LAND	WEIGHT	HEIGHT	AVAIL.	SOURCE
X700	800	800	2896	27 IN.	6 MO.	YES
CS4656-77 (FLAT X-700)	300	500	2896	18 IN.	14 MO.	NO**
CD850-6A	650	650	2995	40 IN.	PROD.	YES
X300	340	340	1333	30 IN.	12 MO.	YES
XTG-411	600	600	2350	26 IN.	PROD.	YES
XTG-250	250	250	1300	29 IN.	6 MO.	YES
X500	550	550	2000	34 IN.	18 MO.	YES
GS400 (STEER ONLY)	216	216	1286	22 IN.	PROD.	YES
X700 TYPE	1320	1320	4700	27 IN.	16 MO.	NO**
CS4656-87 (FLAT X-600)	760	340	1860	18 IN.	14 MO.	NO**
XT1400	750	750	6350*		OUT OF PROD.	YES

* INCLUDES FINAL DRIVES.

** THE ORDNANCE PARTS USED IN THE TRANSMISSION ARE MULTI-SOURCE BUT ALL OTHER PARTS WOULD NOT BE UNLESS SO AGREED BY ALLISON DIV. GMC.

Figure 8-16 Allison Transmissions Data

8.4.1.2 <u>Stratos.</u> The Stratos Division of the Fairchild-Hiller Corporation, in addition to developing hydrostatic elements for truck transmissions, have designed and built, under U.S. Army sponsorship, regenerative steering transmissions for tracked vehicles. They have prepared a concept of a steering unit for the LVTPX12 and pertinent data are shown in Figure 8-17. Another truck-type transmission would be required to provide sufficient ratios for land performance. This proprietary steering unit could not be multi-sourced.

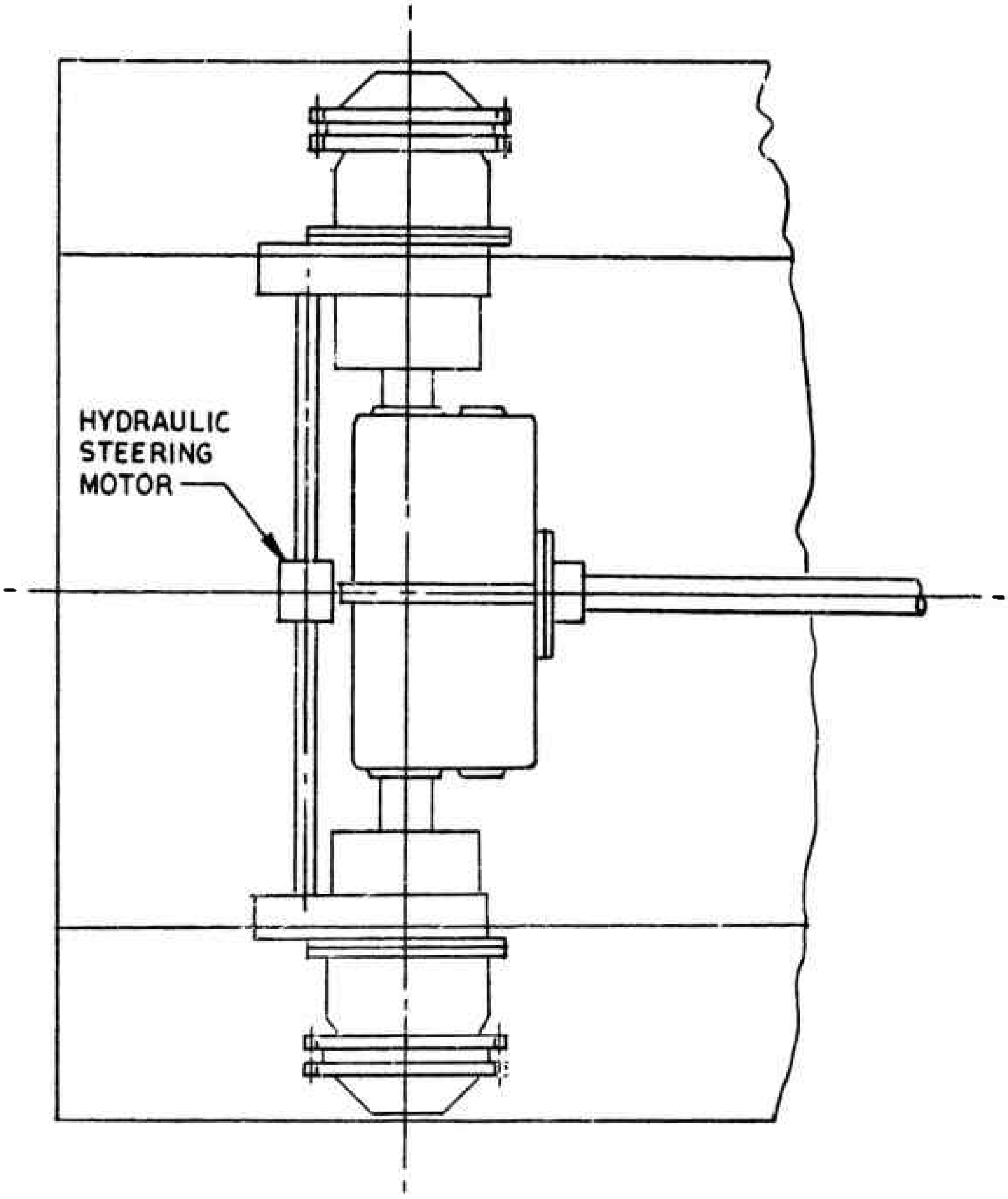
MODEL	INPUT HP WATER	LAND	DRY WEIGHT	HEIGHT	PROTO. AVAIL.
STEERING UNIT	760	340	1250	15 IN.	12 MO.

Figure 8-17 Stratos Steering Unit Data

8.4.1.3 Buehler. Buehler Division of the Indiana Gear Works prepared a concept of a new transmission suitable for the LVTPX12. It features a hydrostatic cross-shaft steer system together with a planetary gear final drive. The concept provides for two outputs from each side of the transmission, one for the drive and the other for the steering. The steer torque is added (or subtracted) from the drive at the combining planetary in each final drive. The transmission would have a maximum height of 18 inches. They estimated that the transmission and final drives less torque converter would weigh 2200 pounds. The design is not far enough along so that a space claiming drawing could be furnished. This design would be proprietary. A sketch of the Buehler concept is shown on Figure 8-18.

8.4.1.4 Goodrich Multitorq. An interesting approach in transmission components is the Multitorq drive unit of the B. F. Goodrich Aviation Products, a Division of the B. F. Goodrich Company. This is a 4-speed, counter-shaft gear, mechanical transmission designed to fit in a wheel. The maximum input power is 200 horsepower at 3800 RPM. For certain concepts, it would be possible to install this within the drive sprockets and have a steering unit with forward and reverse inside the vehicle. The design is proprietary.

8.4.1.5 Others. Other firms have components such as torque converters, ratio changing units, clutches, etc., but there are no steering units



FINAL DRIVE 3.9:1

Figure 8-18 Buehler Transmission Concept

suitable for high speed tracked vehicles. Some suppliers in this category are Rockwell Standard, Clark Equipment, Caterpillar, Long Division of Borg-Warner Corporation, Allis-Chalmers, and Dana.

B.4.2 Hydrostatic Transmissions. Hydrostatic drives have been under development for many years. Currently, they are being used in specialized equipment such as aircraft tow tractors, small garden tractors, industrial materials handling equipment, and special equipment. In most applications the pumps and motors are separated. A newer form has the pumps and motors attached to the same block, allowing much higher pressures to be used with a resultant increase in efficiency. A number of firms are active in the field: Sundstrand Aviation, Stratos Division of Fairchild Hiller, Dynex, Gar Wood, Vickers, General Electric, and Consolidated Diesel. The principle is applied in different ways by individual firms. The two basic approaches are:

- Full hydrostatic
- Hydromechanical

The full hydrostatic systems transmit all of the power through oil under pressure, supplied by hydraulic motors at the wheels for remote drive or by combining the pump(s) and motor(s) into one housing for unit drive. The speed range may be accomplished entirely by hydraulic means, or may utilize gears to give more than one range. Range control is set so that throttle position and output speed will control engine speed at or near the optimum specific fuel consumption point for any vehicle speed or load.

In the hydromechanical approach, oil under pressure transmits a portion of the power while another portion is transmitted mechanically. Efficiency of this approach is somewhat better than the full hydrostatic. Engine speed may be controlled to insure optimum specific fuel consumption for all operating conditions. Hydrostatic steering is employed in both approaches.

8.4.2.1 Sundstrand. Sundstrand Aviation Division of the Sundstrand Corporation is a leading supplier of aircraft constant-speed drives and has diversified into vehicle transmissions. They are supplying small (15 HP) hydrostatic drives for garden tractors and have a contract from ERDL to design and build two transmissions and install one unit in a prototype Universal Engineer Tractor. The installation is due to be completed by mid-1965.

Sundstrand has concepted a hydrostatic transmission for the LVTPX12 that will satisfy the performance requirements and fit under the floor. The transmission consists of two basic systems: the propulsion system which contains a hydraulic pump, and two hydraulic motors back to back with one output to a single power shift; and the steering system made up of one hydraulic pump and one hydraulic motor. The steer system is connected to the propulsion system through a gear differential. Transfer of high-pressure fluid between the related hydraulic units is through one common port plate, eliminating the need for high-pressure lines and fittings. All elements of the transmission, with the exception of brakes and final drives, are integrated into one unit for installation as a single piece of equipment in the vehicle. External air-cooled brakes are incorporated. The only external oil lines required are low-pressure connections to and from the oil cooler. The schematic of the transmission is shown in Figure 8-19.

The power shift device utilized at the output of the propulsion motor in the transmission is a commercially available, hydraulically actuated, high/low-ratio clutch which has been used by Sundstrand for a present hydrostatic transmission application.

The schematic arrangement of the controls is shown in Figure 8-19. The only controls required are the conventional accelerator pedal, brake pedal, a mode selector (forward, neutral, reverse), and a steering wheel. The transmission is fully reversible and can propel the vehicle as fast in reverse as in the forward direction. It also provides the vehicle with a capability of being pushed to start. The operator need only place the mode selector lever in the appropriate position, either forward or reverse, and depress the accelerator pedal.

The use of conventional disc-type brakes, along with the capability of utilizing engine drag, is considered the least complex, most economical, and most efficient braking means. Dynamic braking within the transmission has been discarded due to added complexity of control necessary to provide synchronism and the probably upsizing of the heat exchanger and cooling circuit. Additional engine braking similar to that gained by down-shifting with a conventional gear-type transmission is also provided by placing the mode selection lever in the position opposite that which the vehicle is travelling and depressing the accelerator pedal.

The steering wheel is directly connected to the flow control of the steer pump. When the steering wheel is in the neutral position, the steering pump is at zero displacement, the steer motor is stationary, and the vehicle travels in

a straight line. The flow from the steer pump in either direction is a function of the steering wheel. When the steer motor rotates, it increases the speed of one propulsion motor and reduces the speed of the other through the action of a differential. Thus, at any given speed, the position of the steering wheel determines the difference in speed between the two vehicle tracks. For a certain position of the wheel, this difference in track speed is the same whether the vehicle is stopped (spin turn), or at full speed. Thus, at slow vehicle speed, a given position of the wheel results in a short radius turn, and at high vehicle speed, the same position of the wheel results in a long radius turn. This results in a steering system response similar to that in a normal wheeled vehicle.

The hydrostatic transmission has another advantage in that, because the pump is only required to make up the leakage losses at vehicle stall, it is possible to produce full torque at stall without bringing the engine to its full power. More details on this transmission are provided in Appendix D.

Since the HST-801 exists in preliminary design concepts only, it can be built larger or smaller to suit the vehicle. The effect of input power and vehicle weight on the transmission dry weight are shown in Figure 8-20. For the auxiliary propelled vehicle, not needing high power, the HST-801 could be built in a lower capacity unit, saving weight. This is shown in Figure 8-21, as HST-801A.

The Sundstrand transmissions are proprietary and cannot be multi-sourced without the government purchasing a license. Contract arrangements for the Universal Engineer Tractor (UET) transmissions have been completed, whereby

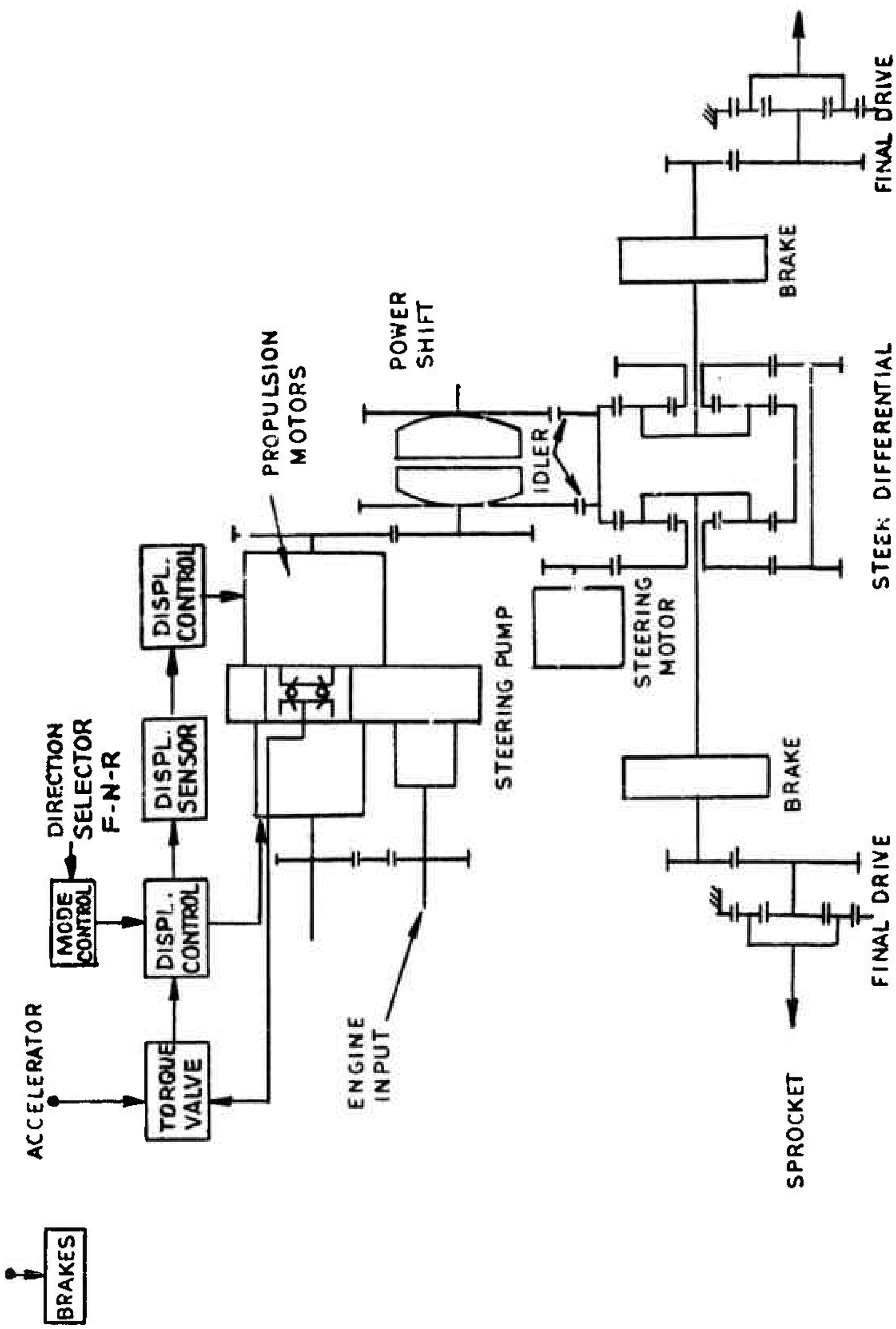


Figure 8-19 Sundstrand HST-8C1 Transmission Schematic

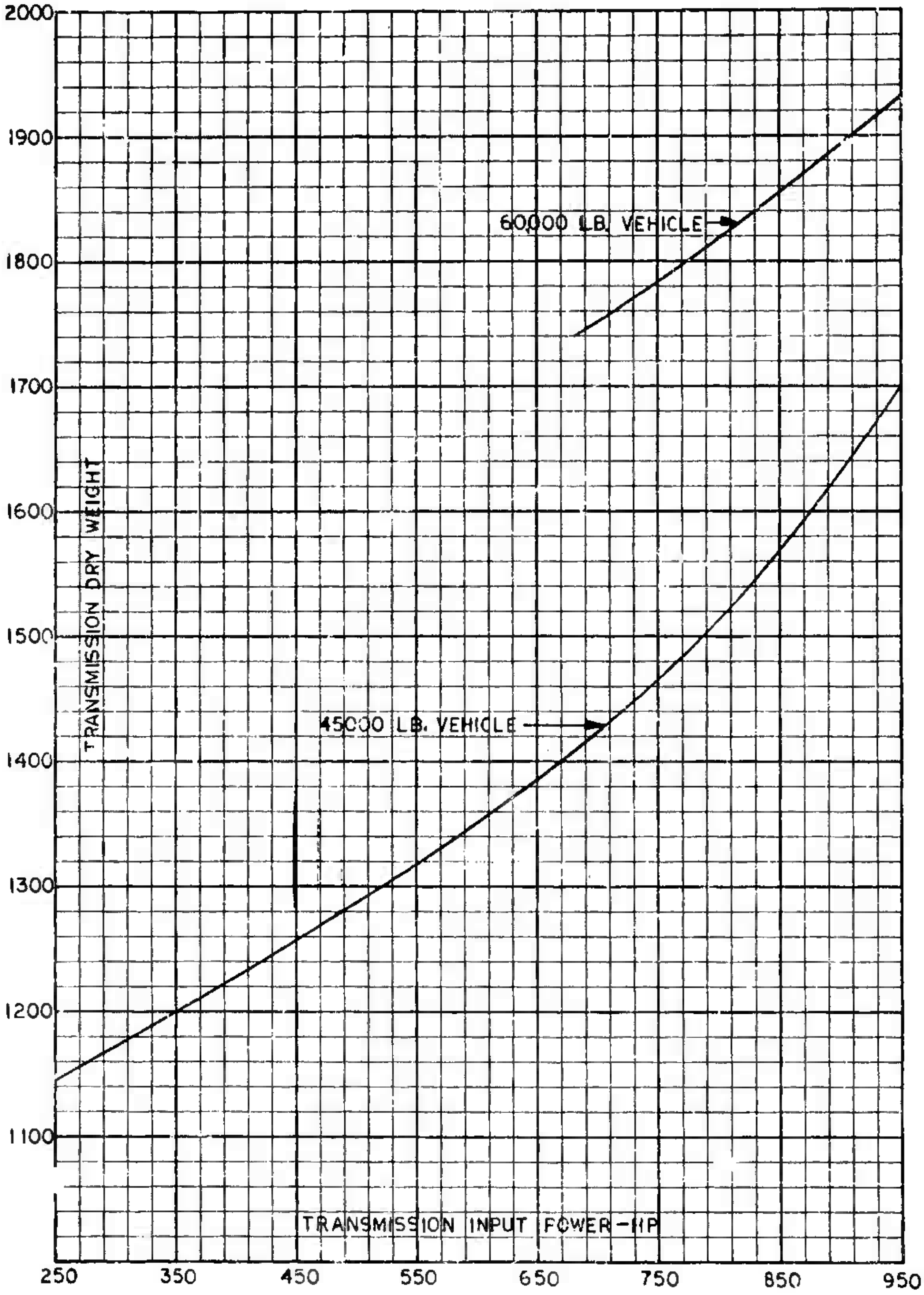


Figure 8-20 Sundstrand HST-801 Hydrostatic Transmission Weight Curve

the government has the option to buy the rights to the data and the design. The govarnmant contracted to purchase two hydrostatic transmissions for tha UET, and at tha same time was givan an cption to purchase the rights to tha data, the dasign, and also, a licanse to prior patents for a fixed price specified in the contract. The price was set as the amount of money Sundstrend has spent on the devalopment, what it will spend of its own money, end profit lost dua to the loss of the data. The government hes until two yaars after the completion of the contract to exercise its option. This covers the UET trensmission only. Within tha same contract, rights to additional data packagas for similer transmissions are also covared. The government has the option to buy rights for any additional transmissions it wishes any time within tan years from the start of the contrect, which is about five to six years aftar the rights to the UET transmission have axpired. The prica for thosa additional data packages is not specifiad, but the machinery has baen established and is: (1) the government would buy the first packaga containing the UET transmission, (2) for the second data package, they would pay for the edditional davelopment raquired, the expansas of preparing the deta package, and royelty payments for Sundstrand patents involved.

Thera is no reason why this arrangement could not ba followad for a transmission for the LVTPX12, and if so, this transmission then would ba e multi-source item.

Tha transmissions availeble from Sundstrend are shown in Figure 8-21.

MODEL	INPUT HP	DRY WEIGHT (POUNDS)	HEIGHT (INCHES)	PROTO. AVAIL.	MULTI-SOURCE	DEVELOPMENT REQUIRED ?
HST-300	350	1300	32	9 MO.	NO*	NO
HST-800	800	2550	37	12 MO.	NO	YES
HST-801	750	1465	15	12 MO.	NO	YES
HST-801A	350	1200	15	12 MO.	NO	YES

* MECHANICS FOR MULTI-SOURCING EXPLAINED IN TEXT, PARAGRAPH 8.4.2.1.

Figure 8-21 Sundstrand Hydrostatic Transmissions Data

8.4.2.2 Stratos and Vickers. The Stratos Division of the Fairchild-Hiller Corporation has been developing a hydrostatic truck transmission for the past several years. They have proposed a tracked vehicle transmission to ATAC and information on this unit is shown in Figure 8-22. Vickers Corporation has also proposed a hydromechanical steering transmission for MBT use. Their concept promises to be the most efficient hydrostatic transmission, since it uses hydrostatic elements to supplement or boost a mechanical drive. A large portion of the operating range is covered by the mechanical drive elements. Some details on this transmission are also provided in Figure 8-22. Other suppliers such as Gar Wood did not have any equipment they would offer for the LVTPX12.

	POWER CAPACITY HP	DRY WEIGHT (POUNDS)	HEIGHT (INCHES)	PROTO. AVAIL.	MULTI-SOURCE	DEVELOP-MENT REQUIRED
STRATOS	900	3300	24	18 MO.	NO	YES
VICKERS	900	3500	OVER 24	18 MO.	NO	YES

Figure 8-22 Other Hydrostatic Transmissions Data

8.4.3 <u>New Transmission Approaches</u>. The power train requirements for the LVTPX12 are quite different from the usual overland tracked vehicle. The higher the water speed desired, the greater the disparity between land and water requirements. A transmission designed for a land vehicle that has sufficient capacity to handle the LVTPX12 (track propulsion) water power is vastly over designed in the other gear ratios. The transmission would have been designed for a 50-ton vehicle which is twice the weight of the LVTPX12. In addition, a greater steering ratio (as high as 3 to 1) is desired in the water mode to provide good maneuverability. Because weight is a very important factor in minimizing vehicle resistance, it is necessary to pick the power train that will have minimum weight. For the track propulsion vehicle, with rear ramp, a forward mounted engine with transmission in the rear, under the floor, offers a reasonable balance and permits the use of a drive sprocket in the rear, which is preferred for drive.

No existing transmissions either in production or in development could be installed under the floor, therefore, the possibility of using a split path drive made up of standard components was investigated. The first concept envisaged the use of a steering unit sized to take the power required for water propulsion. A truck transmission in parallel with that drive line would provide the ratios needed for land use. A special torque converter would be needed to protect the small truck transmission from inadvertent exposure to high power. The converter would be selected to deliver power not in excess of the transmission capacity and would stall at the governed speed of the engine. The engine governor would automatically decrease the power output of of the engine to that which the converter would accept. All components of this

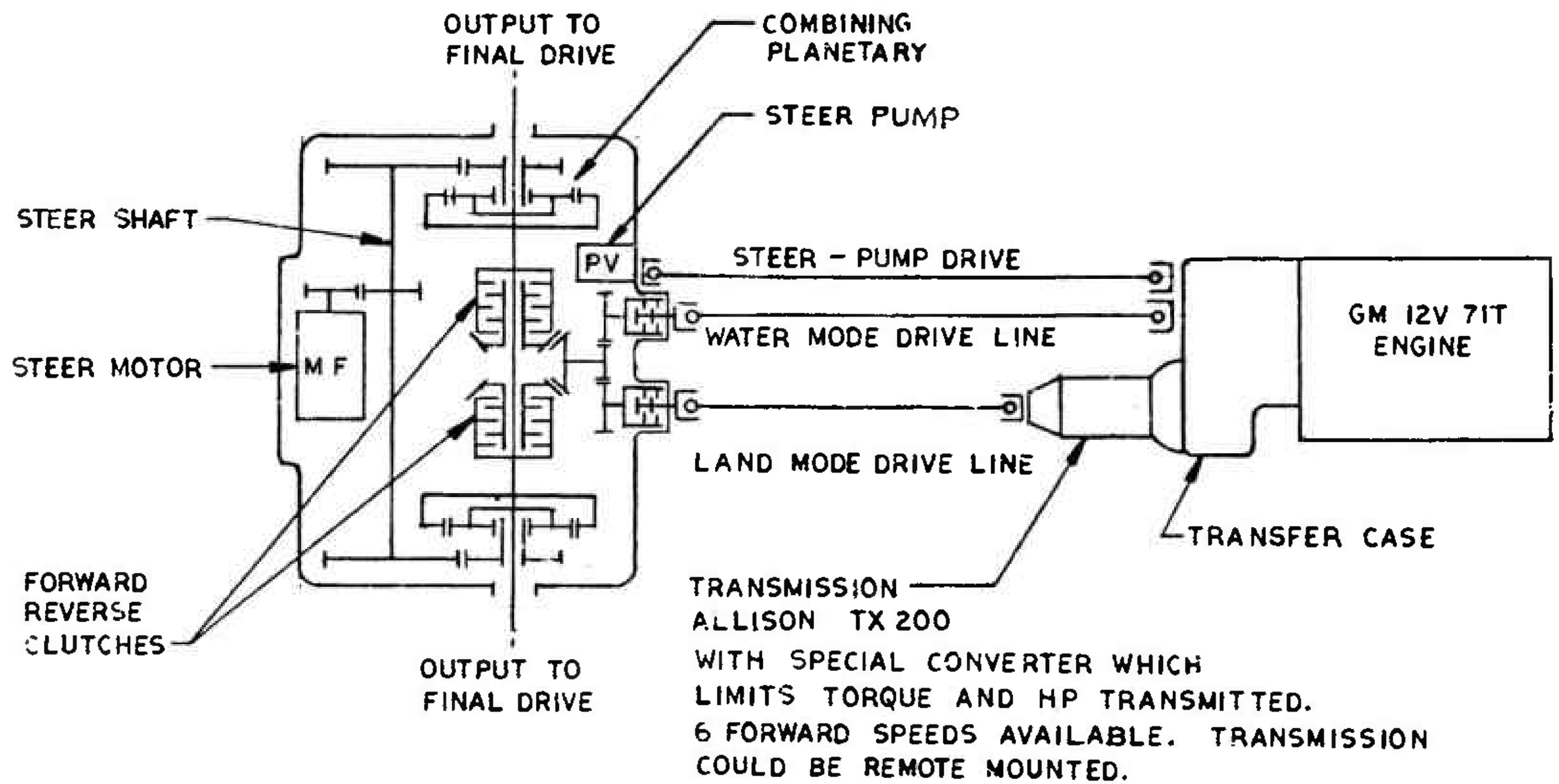


Figure 8-23 Split Path Transmission Concept

system could be multi-sourced. A sketch of this arrangement is shown in Figure 8-23 and more details are in Appendix D.

The split path power arrangement provides several advantages over any existing transmission systems. These are:

- Reduced overall weight
- Use of a truck transmission, TX-200, currently in the military system
- More flexibility in placing transmission components in the vehicle
- Reduced cost over a new development transmission

Further investigation disclosed that the split path approach would be improved by using a planetary gear set at the engine to split the engine torque between the truck transmission and the direct connection to the steering unit. The steering transmission would collect the dual outputs, provide reverse, and incorporate the hydrostatic steering function. In order to provide good steering performance on both land and water, two steering gear ratios are required. Special limiting governors cause automatic upshifting to prevent overspeeding of the transmission gears, and the engine governor will prevent overspeeding in top gear. This arrangement would make use of an existing ratio change transmission and would require the design of the steering section only. Some weight and size reductions are expected for 800 horsepower capacity when compared to the existing CD-850 or the new X-700 units, each of which weigh approximately 3000 pounds. In addition, this arrangement could be multi-sourced. More detail is in Appendix D.

Allison Division of General Motors Corporation studied the effects of the increased inertia loading on the TX-200 transmission and indicated that tests and additional investigation would be required before approval could be given to this application. Sundstrand, Allison, and Stratos were asked to submit concepts on the steering transmission, but Sundstrand and Allison declined with the explanation that the steering unit transmission would be very close to a complete transmission and would offer no gains in weight, size, or cost. Stratos submitted a concept. The concept data are listed in Figure 8-17.

An investigation was conducted to explore the idea of uprating a transmission suitable for land use to handle the power needed for water propulsion. The Allison X-500 transmission could be changed by increasing the strength of components in the second-speed gear train to handle the high power in converter lockup. The torque in lockup is not too much greater than that encountered at lower powers when a converter is used. Allison rearranged the X-500 components so that proper water steering ratios and adequate power absorption capabilities would be provided in second gear. The maximum height of the transmission is 18 inches with most of the unit at 15 inches. This represents a new transmission which uses many components of the X-500. This unit would be a proprietary design, not multi-source. Data on this unit are shown in Figure 8-16.

Continued analysis of the transmission problem resulted in a completely different approach, referred to herein as the Chrysler CCS Concept. All steering transmissions use planetary gear sets in their output to combine the steering torque with the main drive torque. In addition, the final drives of the vehicle provide another reduction. The steering or combining planetary

gear sets can be taken out of the transmission and put in the final drive. The result is a transmission and final drive that weighs about the same as the lightest transmission, the Sundstrand hydrostatic unit, and is 20 percent lighter than the flat X-600 transmission. The concept features countershaft gearing, commercially available hydraulic clutches, commercial torque converter, and available hydraulic elements for the steering. The transmission has four speeds forward and two in reverse. Second gear in lockup converter for the CCS-1 has sufficient capacity to take the full output of the Detroit Diesel 12V71T engine. The CCS-2 unit is slightly smaller because the high power capability in second gear is not needed for the auxiliary-propelled vehicle. The transmission has two outputs on each side, one for the main drive and the other for the steering output. This transmission would be Government-owned and could be multi-sourced. Pertinent details are shown in Figures 8-24 and 8-25.

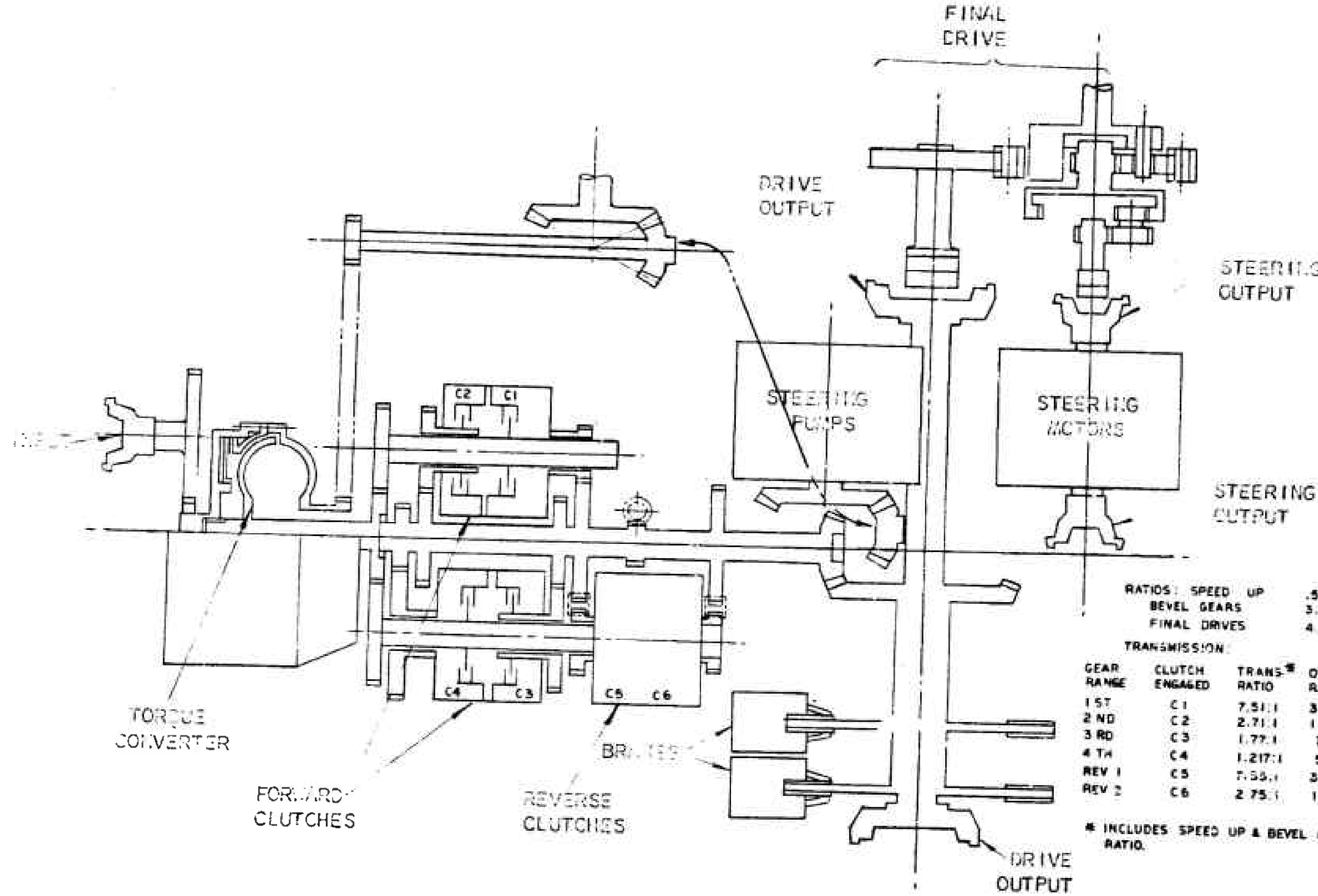
MODEL	INPUT HP WATER	INPUT HP LAND	DRY WEIGHT (POUNDS)	HEIGHT (INCHES)	PROTO. AVAIL.	MULTI-SOURCE
*CCS-1	760	340	1500**	15	12 MO.	YES
CCS-2	340	340	1400**	15	12 MO.	YES

** Including the final drives, (equivalent final drives used with other transmissions weight 300 lb. each)

* CCS stands for "Concept Counter Shaft"

Figure 8-24 Chrysler Concept Transmission Data

8.5 <u>Initial Tradeoff - Drive Systems</u>. There are many transmissions or drive systems that could be used in the LVTPX12 depending on the configuration and requirements to be met. The following guidelines for selection of a drive system are complicated and many are not quantitative.



RATIOS:	SPEED UP		.595:1
	BEVEL GEARS		3.956:1
	FINAL DRIVES		4.21:1

TRANSMISSION:

GEAR RANGE	CLUTCH ENGAGED	TRANS.* RATIO	OVERALL RATIO
1 ST	C1	7.51:1	31.62:1
2 ND	C2	2.71:1	11.41:1
3 RD	C3	1.77:1	7.45:1
4 TH	C4	1.217:1	5.21:1
REV 1	C5	7.55:1	31.79:1
REV 2	C6	2.75:1	11.58:1

* INCLUDES SPEED UP & BEVEL GEAR RATIO.

Figure 8-25 Chrysler CCS Transmission Schematic

- Weight
- Size, particularly height
- Availability of prototypes
- Ease of maintenance
- Complexity of installation
- Power capacity
- Cost, both prototype and production
- Interchangeable with other vehicles
- Amount of development required
- Reliability
- Safety
- Compatibility with the military supply system
- Multi-Source capability
- Braking performance
- Steering performance, both land and in water
- Efficiency
- Adequacy of ratio coverage

The transmission to be selected for the US/FRG Main Battle Tank is of vital significance to the LVTPX12 program. If the tank transmission could be used there would be great savings in time, material, funds, supply support, etc. The only transmission being seriously developed by the United States for the MBT is the Allison X-700. The maximum input power this can take is 825 horsepower or a nominal 800 horsepower. The trend in MBT power is for input power at the 1200 horsepower level or higher. At the time of this writing (April 1965), there has been no announcement of a transmission selection. It is anticipated that it will not be the X-700, although it may be a larger

version of the X-700. It is also probable that a German transmission may be selected. In any event, it will be a transmission designed for a 40-plus ton land vehicle with a power capacity in excess of 1000 horsepower. At this time, it appears that it would be a serious handicap to the LVTPX12 to incorporate a transmission that large.

The following criteria were set up to make an initial selection of drive systems.

- Track Propelled

Transmission Power Capacity*	- at least 700 HP in the water mode
Dry Weight	- 3.6 pounds per HP or less, based on state-of-the-art development
Availability	- prototypes in 14 months or less which is the latest possible time that could meet prototype build schedule

- Auxiliary Propelled

Transmission Power Capacity*	- at least 270 HP in the land mode
Dry Weight	- 3.9 pounds per HP or less
Availability	- prototypes in 14 months or less

* This represents the lowest SHP (or Net Engine horsepower) that could be used, reference Section 4.0 and Appendix D.

The transmissions that meet these criteria are listed in Figures 8-26 and 8-27.

MFR.	MODEL	POWER WATER (HP)	DRY WEIGHT	DRY WEIGHT LB/HP	MULTI-SOURCE	PROTO. AVAIL.
ALLISON	X-700	800	2896	3.6	YES	6 MO.
ALLISON	FLAT X-700	800	2896	3.6	NO	14 MO.
SUNDSTRAND	HST-800	800	2550	3.2	NO	12 MO.
ALLISON	FLAT X-600	760	1860	2.4	NO	14 MO.
	CCS-1	760	1500*	1.0	YES	12 MO.
STRATOS	STRG. UNIT, TX-200 & TRAN. CASE	760	2015**	2.6	NO	12 MO.
SUNDSTRAND	HST-801	73[illegible]	1465	2.0	NO	12 MO.

* The 1500 LB. includes 2 final drives which weigh 300 LB each for other transmissions.

** The Stratos steering unit weighs 1250 LB, the TX-200 transmission 465 LB, and the necessary transfer case at the engine 300 LB.

Figure 8-26 Transmission Candidates for Track-Propelled LVTPX12

8.6 Engine Selection. Paragraph 8.3 listed general guidelines that should be followed for the initial selection of an engine for the LVTPX12. Thirteen of these factors have been selected for the final engine trade-off and they are:

- Cost
- Performance
- Weight
- Space
- Reliability
- Maintainability
- Multi-source

- Multi-fuel
- Complexity of installation
- Availability of prototypes
- Standardization
- Human factors
- Environment

MFR.	MODEL	POWER LAND (HP)	HEIGHT (IN.)	DRY WEIGHT	LB/HP	MULTI-SOURCE	PROTO. AVAIL.
ALLISON	X-700	800	27	2896	3.6	YES	6 MO.
SUNDSTRAND	HST-800	800	37	2550	3.2	NO	12 MO.
SUNDSTRAND	HST-801	750	15	1465	2.0	NO	12 MO.
ALLISON	CD-850	650	40	2995	4.6	YES	PROD.
ALLISON	XTG-411	600	26	2350	3.9	YES	PROD.
ALLISON	FLAT X-700	500	18	2896	5.8	NO	14 MO.
SUNDSTRAND	HST-300	350	32	1300	3.7	NO	9 MO.
SUNDSTRAND	HST-801A	350	15	1200	3.4	NO	12 MO.
ALLISON	X-300	340	30	1333	3.9	YES	12 MO.
ALLISON	FLAT X-600	340	18	1860	5.5	NO	14 MO.
	CCS-1	340	15	1500*	2.4	YES	12 MO.
	CCS-2	340	15	1400*	2.1	YES	12 MO.

* The weight shown includes 2 final drives which weigh 300 LB each for the other transmissions.

Figure 8-27 Transmission Candidates for Auxiliary-Propelled LVTPX12

8.6.1 Engine for Track-Propelled Vehicle. Cost in the trade-off analysis represents all expenses in the program chargeable to the engine for 1000 vehicles over the 15-year life. The categories considered are:

- Design
- Development
- Prototypes (8)
- Spares for 1 year of testing
- Prototype test rig preparation
- Prototype test rig operation
- Tooling
- Pre-production pilot models (25)
- Spares
- Production models (1200)
- Operational costs for 15 years and 1000 vehicles
    - Fuel consumption
    - Oil consumption
    - Servicing
    - Overhauls
    - Repairs

The development of these quantities is in Appendix D, along with the cost for an ideal engine. The ideal engine is assumed to be currently in production for another tracked vehicle. As a consequence, the design, development, prototype test, and tooling would all have zero cost. The fuel consumption was based on a liquid-cooled engine having a brake specific fuel consumption

of 0.4 lbs per brake horsepower hour. The 15-year cost for the ideal engine is 25.8 million dollars and is given the score of 10. For each three million dollars more that the subject engine cost, 1 point is deducted

The next most important factor is engine Performance as determined by the water speed of the vehicle. The water speed specification is 8 miles per hour required and 10 miles per hour desired. If the engine developed enough horsepower at the track to drive the vehicle at 10 miles per hour, it is rated 10. For each 0.4 MPH difference from this, 1 point is added or subtracted, depending on whether the vehicle is faster or slower than 10 miles per hour. The derivation of the values are shown in the calculations in Appendix D.

The third factor is Weight. The engine that meets all the other requirements and is the lightest would obviously be the best engine for the LVTPX12. The weight of the engine, cooling system, fuel and fuel system is used. A total system weight of 6000 lbs is taken as the baseline and 1 point subtracted for each 500 pounds over or 1 point added for each 500 pounds under this baseline figure.

In order to make a light, speedy vehicle and still provide adequate space for personnel and cargo, the smallest possible Space should be occupied by the engine and its associated systems. The properties of engine length and total engine system volume are the criteria. A short engine permits the gunner to stand on the floor of the vehicle in front of the engine. Fifty inches length is the desired length and given a rating of 2. Sixty inches is given a rating

of I and lengths differing from these are interpolated or extrapolated. The engine volume, cooling system volume and fuel volume are all added together. The base volume is taken as 100 cubic feet and given a score of 8. The poorest is 200 cubic feet and given a score of 0.

There are six levels of Reliability considered; the first, for engines in production, scores 10. The second, prototypes built and similar to engines in production, components in extensive use, would score 8. If prototypes were built and tested, the score is 6. If only prototypes were built and testing not complete, the score is 4. If there is a design only, the score is 2, and for a concept only, the score is 1.

Maintainability was judged by the ease of maintenance, the amount of parts in the government supply system, and the knowledge of mechanics in the government service. Ease of maintenance is scored 2 for the ideal engine. If all the parts of the engine are in the government supply system, it would score 4. If the mechanics in the government service had extensive experience in repairing this engine, then it would score 4. The total maintainability score is the sum of these three ratings.

Multi-Source is a very important part of the evaluation, and three possibilities are considered. The first, the government owns the entire design, a yes answer here scores 10. Two, the government owns the entire design except for certain commercial components which are envelope drawings, a yes answer to this scores 8. Third, the government has a license to use those items not covered by either the first or second above, a yes answer here scores 6.

The Multi-fuel capability was evaluated by rating each engine on how well it would burn the four fuels; diesel (MIL-F-16684), CIE (MIL-F-45121), JP5 (MIL-F-7914), and gasoline (MIL-G-3056), listed in the specification. The ability to burn the fuel with full power and full durability is given a score of 2.5. The ability to burn fuel with full durability but some loss in power is scored 2. The ability to burn fuel with a loss of power and durability is scored 1.5. The ability to burn a fuel only in an emergency basis with possible malfunctioning or failure is given a score of 0.8.

The Complexity of the installation was judged on the basis that 10 was ideal.

The Availability of prototypes is a crucial criterion but this was also one of the factors used in the initial trade-off, and for that reason, this factor has been given less importance now. The engines are scored on the basis that 12-month availability scores 2, 9-month availability scores 6, and 6-month availability scores 10.

If the engine for the LVTPX12 is being used in another tracked vehicle, or could be used in another tracked vehicle, it has a decided advantage (Standardization) over its competitors. If the engine is in present military use, it scores 2.5; if components are in use, it scores a maximum of 2.5 and the score is proportioned down according to the amount of components in present use. If the engine is released for future use, it scores 2.5, otherwise it scores according to its chances at this moment. Future use of components is also scored in the same fashion with a maximum of 2.5 if the components are released for use, and scaling off depending on the chances of components being released for use.

To evaluate human factors; noise, vibration, fumes, and safety are considered with each one having a maximum score of 2.5.

The three major elements of the environment chosen for Sensitivity to Environment evaluation are salt water, humidity, and temperature. If the engine is insensitive to salt water, it scores 4. If the engine is insensitive to humidity, it scores another 3. If it is insensitive to temperature, it also scores another 3 for a maximum of 10.

Each of the features rated have a maximum score of 10, but all features do not have the same value to the vehicle and program. Accordingly, a weighting factor is applied to the individual scores corresponding to the value of the characteristic to the vehicle. Cost is considered the most important and is given the weighting factor of 20. Performance and Weight are next and are weighted 15. Space is given the weight of 10. The others are weighted either 4 or 5 according to their importance. The weightings are shown on the summary chart, Figure 8-28.

8.6.2 Track-Propelled Engine Selection. The two engines remaining after the initial selection are the Detroit Diesel 12V71T and the Continental AVDS-1100 VCR engines. Using the criteria explained in the previous paragraphs, the 12V71T engine is outstanding in Weight, Cost, Space, and Maintainability, and deficient in Performance and Multi-Sourcing. Overall, its score is much higher than the AVDS-1100 VCR engine; therefore, the 12V71T engine is recommended for the track-propelled version of the LVTPX12. Figure 8-28 is a summary of the ratings. The breakdown of the ratings for each engine, along with calculations of performance and weights for other

engines are in Appendix D. Figures 8-29 and 8-30 are power curves and fuel maps for the two engines.

	WEIGHTING FACTORS		12V71T RATING	12V71T SCORE	AVDS-1100 VCR RATING	AVDS-1100 VCR SCORE
1	COST	20	5.6	112	3.8	76
	MILLIONS OF DOLLARS		38.9		44.3	
2	PERFORMANCE	15	5.2	78	6.0	90
	DHP		637		808	
	WATER SPEED		8.1		8.4	
3	WEIGHT	15	9.4	141	6.6	99
	POUNDS		6298		7727	
4	SPACE	10	6.0	60	3.8	38
	LENGTH (INCHES)		54		58	
	VOLUME (CU.FT.)		145		168	
5	RELIABILITY	5	8	40	6	30
6	MAINTAINABILITY	5	7.5	38	3.5	18
7	MULTI-SOURCE	5	0	0	6	30
8	MULTI-FUEL*	5	9.0	45	9.0	45
9	COMPLEXITY	4	8	32	7	28
10	AVAILABILITY	4	10	40	10	40
11	STANDARDIZATION	4	3.5	14	3.0	12
12	HUMAN FACTORS	4	9.5	38	9.0	36
13	ENVIRONMENT	4	6.5	26	6.5	26
	TOTAL	100		664		568

*(Both engines receive a score of 2.5 for all fuels except gasoline. For gasoline the score is 1.5)

Figure 8-28 Track-Propelled LVTPX12 Engine Trade-Off

CHRYSLER CORPORATION

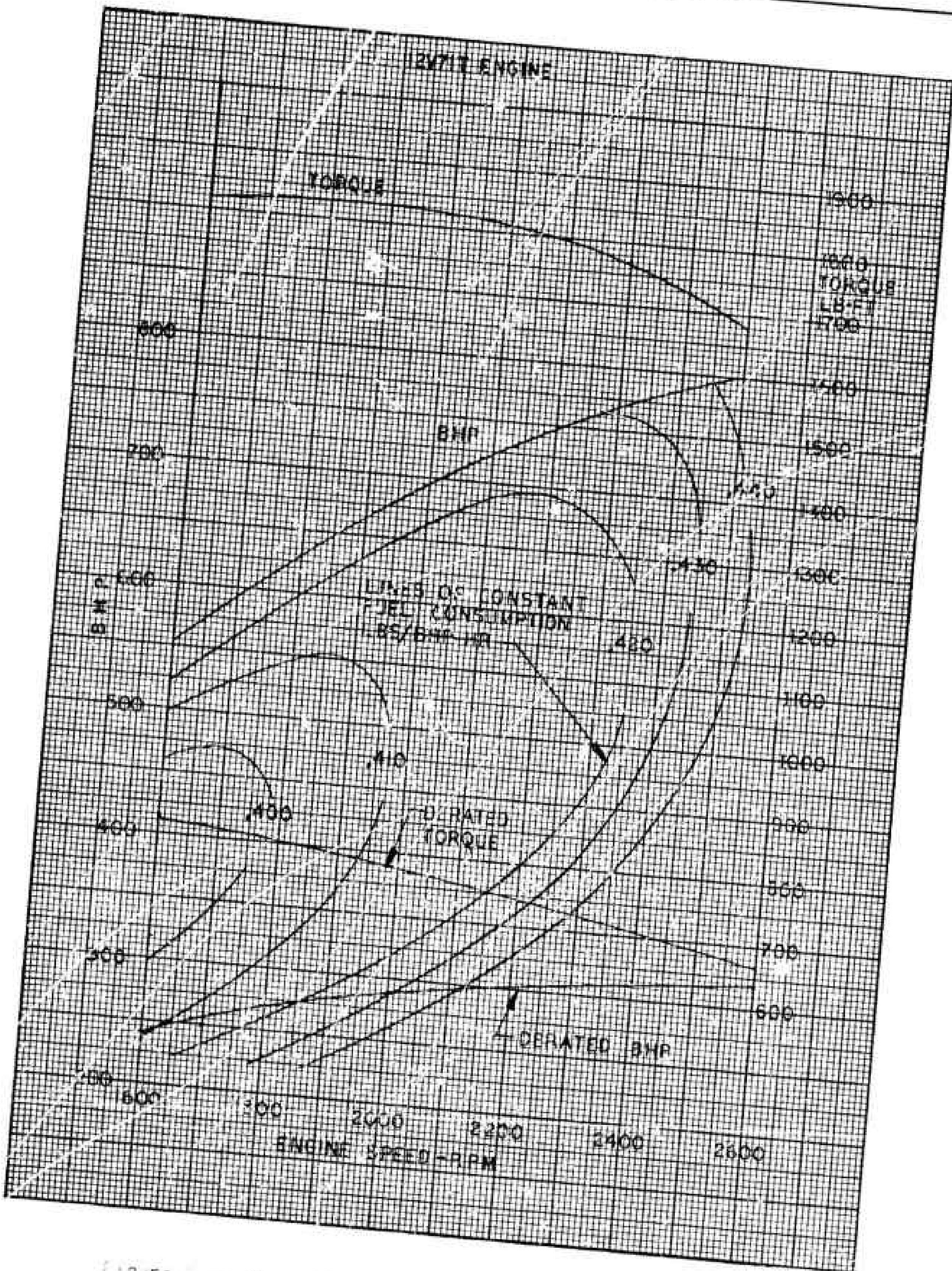
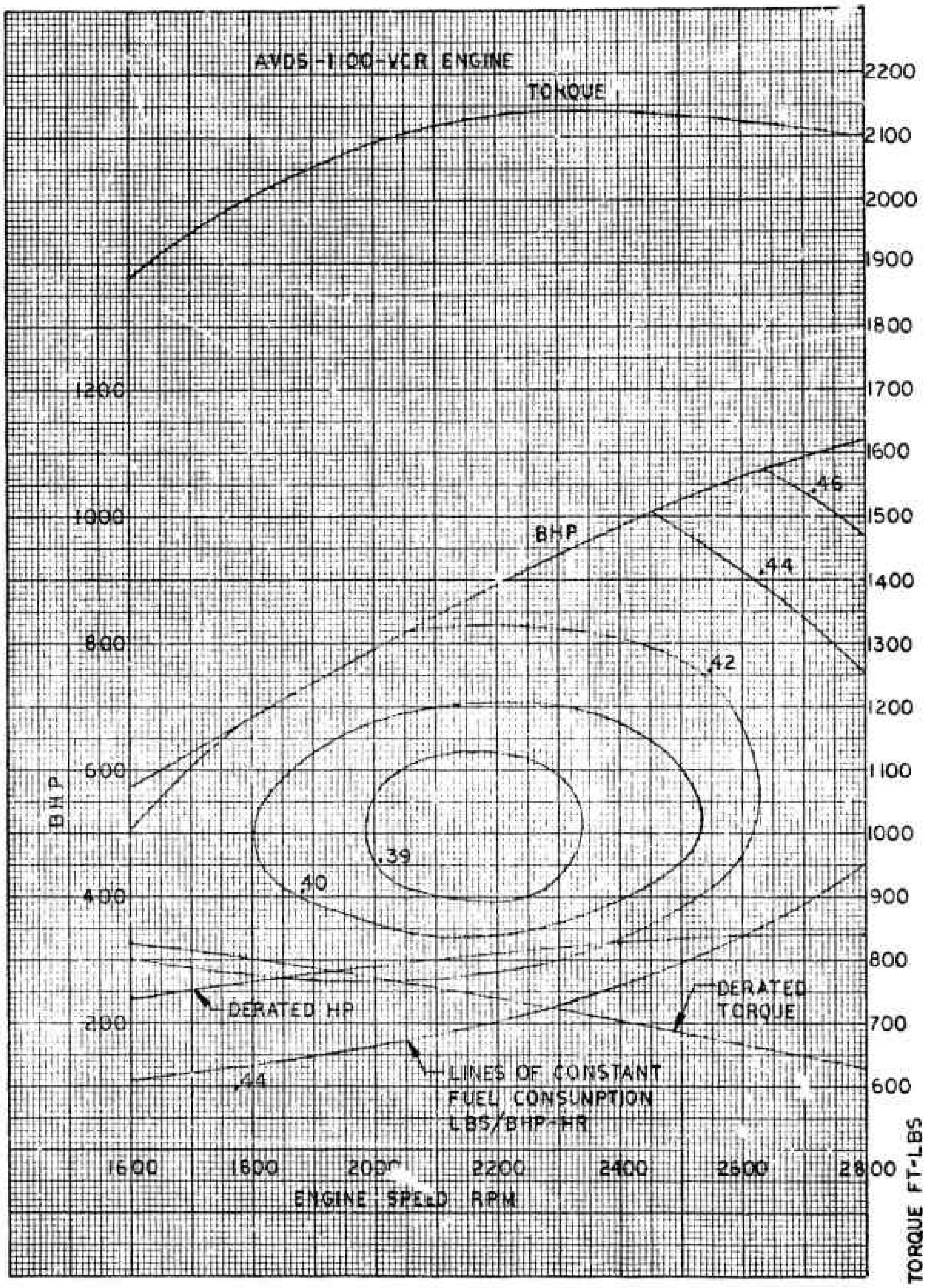


Figure 8-. Power Curve and Fuel Map for 12V71T Engine


AVDS-1100-VCR ENGINE
TORQUE
BHP
.46
.44
.42
.40
.39
.44
DERATED HP
DERATED TORQUE
LINES OF CONSTANT FUEL CONSUMPTION LBS/BHP-HR
1200
1000
800
600
400
200
BHP
2200
2100
2000
1900
1800
1700
1600
1500
1400
1300
1200
1100
1000
900
800
700
600
1600
1800
2200
2400
2600
2800
ENGINE SPEED RPM
TORQUE FT-LBS

A General Motors diesel engine is admittedly a problem when viewed from a multi-source standpoint. These engines are designed, tested, and developed without benefit of government funding. Under these conditions, the engines are proprietary and design drawings are guarded by General Motors. Without a complete detailed set of design drawings, the government cannot obtain either new engines or repair parts from any source other than General Motors. ASPR and multi-source regulations cannot be followed without free use of the detailed design drawings.

This poses two problems: the first one is major procurement of these engines for new production or procurement during national emergency and is restricted to one source. The second item is replenishment procurement of repair parts for vehicle support and is similarly limited to one source.

From a practical standpoint, no problem in either engine procurement or repair-parts procurement has been brought to the attention of this contractor. Several informal telephone calls were made to check this multi-source problem. One check was with Marine Corps Supply Agency who reported unofficially that they experienced no difficulty in either the provisioning cycle or in subsequent repair-parts replenishment action. It was also learned that they are well satisfied with the service on their procurement and provisioning actions and rarely experienced difficulty in repair-part quality. Such a situatic is not always evident when procurements are made from other than the original manufacturer. MCSA also reported that G.M. diese.s have a wide use in the U. S. Marine Corps with additional use being accepted at the present time.

Also, as a multi-source item, the main power plant is a controversial item, while it is true that a government-owned design of an engine may be manufactured by other than the original manufacturer, it is not a common occurrence and is expensive because of experience and tooling. If one considers the engine assembly as a complete unit with the multi-source requirements being satisfied by exchange with another engine of equal performance, then duplication expense is prohibitive. An LVTPX12 that could incorporate two different engines would require: (1) Varying duplication of design for cooling, electrical, mounting, (2) duplicate provisioning and instruction manuals, and (3) duplicate stocking of repair parts.

In summary, this contractor believes that the proprietary status of General Motors diesels will pose no practical problems in supporting the engines in the LVTPX12.

8.6.3 Auxiliary-Propelled Vehicle Criteria. The criteria for the auxiliary-propelled vehicle are basically the same as for the track-propelled vehicle with the only differences being in Cost, Performance, Weight, and Space. The propellers drive the vehicle faster through the water for the same power giving the engines a higher rating. Less power is required to drive the vehicle at 8 MPH on water and this reduces the fuel requirements showing up in the other three factors.

8.6.4 Auxiliary-Propelled Vehicle Engine Selection. There are 3 engines left after the Initial Selection is made and these are:

- Detroit Diesel 12V71T
- Continental AVDS-1100 VCR
- Caterpillar LVMS-1050 VHO

The 12V71T engine showed up best in Cost, Weight, Reliebility, end Maintainability; the AVDS-1100 VCR engine was best in Performance; and the LVMS-1050 VHO engine was best in Space, Multi-Source, and Multi-Fuel. Overall, the 12V71T engine has the best rating and is the engine recommended for the auxiliery-propelled LVTPX12.

The summery of ratings are in Figure 8-31 and the details on the calculations of the retings ere in Appendix D. There are also calculetions of performence end weights for other engines for comparison purposes in Appendix D. Figures 8-29, 8-30 and 8-32 are the power curves for the engines.

8.7 Transmission Selection. With the selection of the engine, another screening of transmissions can be made before the final selection.

8.7.1 Interim Selection Track-Propelled Vehicle Transmissions. Comparisons of the power delivered by the transmissions in Figure 8-26 as shown in the calculations in Appendix D, disclose that the hydrostatic transmission, HST-801, has losses much greater than those of the other transmissions. This trensmission, together with the 12V71T engine, does not supply enough power to the track to propel the vehicle at 8 MPH. Therefore, the HST-801 transmission is deleted from the final trade-off analysis. Comparison of the Stratos split-path approach and the others also indicates that this transmission should be dropped before the final trade-off. The steering transmission would require almost es much development as a new trensmission or the flat X-600. The flat X-700 has very little advantage over the flat X-600 to justify the extra 1000 pounds and so it is dropped. The HST-800 is dropped for the same reason beceuse it offers very little advantage over the HST-801. In addition, the vehicle concept requires the transmission to be under the floor in the stern and the

WEIGHTING FACTORS		12V71T RATING	12V71T SCORE	AVDS 1100 RATING	AVDS 1100 SCORE	LVMS 1050 RATING	LVMS 1050 SCORE
1 COST	20	5.7	114	3.9	78	2.5	50
MILLIONS OF DOLLARS		38.6		44.0		48.0	
2 PERFORMANCE	15	11.8	177	12.5	187	11.2	168
OHP		680		846		609	
WATER SPEEO		10.7		11.0		10.5	
3 WEIGHT	15	12.1	182	10.6	159	11.5	172
POUNOS		4969		5712		5238	
4 SPACE	10	8	80	6.7	67	10	100
LENGTH		54		58		51	
VOLUME		120		131		99	
5 RELIABILITY	5	8	40	6	30	4	20
6 MAINTAINABILITY	5	7.5	38	3.5	18	3.3	16
7 MULTI-SOURCE	5	0	0	6	30	8	40
8 MULTI-FUEL	5	9.0	45	9.0	45	9.5	48
		1.5*		1.5*		2.0*	
9 COMPLEXITY	4	8	32	7	28	7.5	30
10 AVAILABILITY	4	10	40	10	40	6	24
11 STANDARDIZATION	4	3.5	14	5.0	12	[illegible]	7
12 HUMAN FACTORS	4	9.5	38	9.0	36	9.0	36
13 ENVIRONMENTAL	4	6.5	26	6.5	26	6.5	26
TOTAL			826		756		737

(* Gasoline rating shown, all other fuels rated 2.5)

Figure 8-31 Auxillary-Propelled LVTPX12 Engine Trade-Off

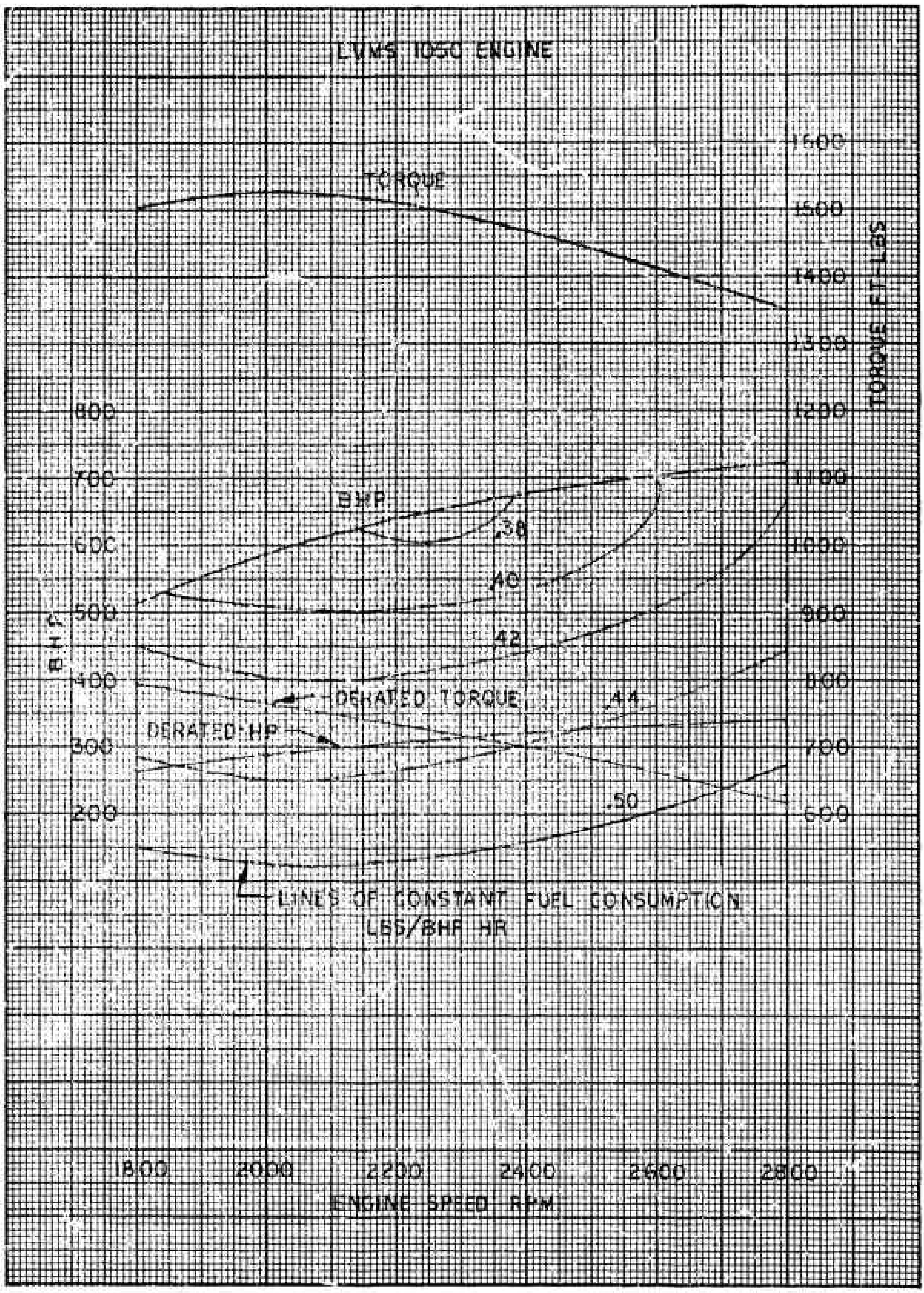


Figure 8-32 Power Curve and Fuel Map For LVMS-1050 Engine

maximum allowable height is 18 inches (Refer to Section 5.0). This criterion eliminates the Allison X-700 transmission. The two remaining transmissions are:

- Allison Flat X-600
- Chrysler Concept CCS-1

8.7.2 Interim Selection Auxiliary-Propelled Vehicle Transmissions. Comparisons of the transmissions initially selected for the auxiliary-propelled vehicle, Figure 8-27, shows that a few units can be dropped because they offer no advantages over other transmissions in the list. The HST-800 and HST-801 are bigger and heavier than the HST-801A which is entirely adequate for the application. The flat X-700 is dropped because the flat X-600 is adequate and the CCS-1 is dropped because the CCS-2 is adequate. The transmissions remaining are:

- Allison X-700
- Allison CD-850
- Allison XTG-411
- Allison X-300
- Allison Flat X-600
- Sundstrand HST-801A
- Sundstrand HST-300
- Chrysler Concept CCS-2

The vehicle concept requires that the transmission be put in the stern under the floor (Refer to Section 5.0). In order to keep the height of the vehicle

and the cargo compartment within the specifications a maximum height of 18 Inches la placed on the transmisaion. When this restriction is imposed, the following transmisaions are left:

- Allison Flat X-600
- Sundstrand HST-801A
- Chrysler Concept CCS-2

8.7.3 Track-Propelled Vehicle Transmission Selection Criteria. A format similar to that employed for the engines is used to make the final selection for transmissions. Fifteen factors from the liat in Paragraph 8.5 are evaluated. They are:

- Cost for the complete program including 15 years of operation
- Efficiency which is a measure of power delivered to the track and fuel consumed
- Weight of transmission, oil, cooling system, and final drives
- Steering performance
- Space which includes the height of transmission and volume of transmission plus final drives
- Adequacy of ratio coverage for both land and water
- Reliability
- Braking performance
- Availability of prototypes
- Maintainability
- Standardization
- Complexity

- Multi-source
- Human Factors
- Sensitivity to environment

The determination of Cost was done in a similar manner to that which is used for the engine. The fuel cost calculated is based on the power loss in the transmission using the 12V71T engine specific fuel consumptions for the actual and ideal transmissions. The cost baseline as figured for an ideal transmission is $23,900,000 and scores 10. One point is deducted for each $4,000,000 more expense charged to that transmission.

Efficiency is rated by the amount of power loss in the transmission in the water mode gear and whether or not the transmission has a lockup for the torque converter. The baseline power loss is 40 horsepower which rates 8. One point is deducted for each additional 5 HP loss. If the transmission has a lockup clutch for the torque converter, then 2 additional points are given.

The Weight factor rates the transmission with oil, cooling system (up to the heat exchanger on the engine), and the final drives. The final drives are included because the CCS-1 transmission has the final drives functionally integrated with the transmission and it would be very difficult to separate the two for an evaluation. The baseline transmission weight is 1,500 pounds and 1 point is deducted for each 400 pounds greater weight.

Five aspects of Steering are considered: If the transmission has fully modulated steering such as hydrostatic steering, 2 points are given; if the steering ratio on land in high gear is 1.4, then 2 points are given, and

if not 1 point is deducted for each 0.2 units difference. Three points are given if the steering ratio in the water mode is 2.5 and 1 point is deducted for each 0.3 unit difference. Pivot steer in neutral capability is given 2 points and an additional 1 point is given if a true pivot about the center of the vehicle is provided.

One of the very critical parts of Space claiming for the transmission is its height. The maximum height that can fit under the floor is 18 inches and 15 inches is desired. A score of 7 points is given to the transmission with a 15 inch height and 1 point is deducted for each one inch increase. The volume of the transmission and final drives would give a maximum rating of 3 if the volume were 10 cubic feet. One point is deducted for each 10 cubic foot increase in volume. Once the height is met any increase in volume only decreases the space available for fuel or other components.

The adequacy of Ratio Coverage was rated by whether or not the following vehicle requirements could be met with the subject transmission and the 12V71T engine.

- Land-top speed of 36 MPH
- Land-speed of 2.5 MPH on the 60 percent slope
- Land-speed of 2.5 MPH on the 60 percent slope in reverse and 8 MPH on the level
- Water - 8 MPH forward
- Water - 3.5 MPH reverse

A yes answer to any of these gave the transmission 2 points.

The measure of Reliability is very similar to that employed for the engines but with the scale extended.

- Transmission in production — scores 10
- Prototypes built, tested; similar transmissions in production; components in extensive use. — scores 8
- Prototypes built and tested — scores 6
- Prototypes built — scores 4
- Design only — scores 2
- Concept only — scores 1

Four aspects of Braking performance are rated.

- Able to hold on 60 percent slope — scores 3
- Deceleration rate of at least 0.5 g — scores 3
- Retarders incorporated — scores 2
- Sensitivity to environment, insensitive — scores 2

The maximum score is 10.

Availability, Maintainability, Standardization, Complexity of Installation, Multi-source, Human Factors, and Sensitivity to Environment are rated the same way as for the engines.

Each of the features rated have a maximum score of 10, but all features do not have the same value to the vehicle. A weighting factor is applied according to the value of the feature to the vehicle. Cost has the greatest

impact and was given the value of 16. Efficiency and Weight are next in importance and given a weight of 11. The remaining features are weighted in a similar manner and the values are shown on Figure 8-33.

8.7.4 Final Selection - Transmission for Track-Propelled Vehicle. Using the above rating method, the CCS-I transmission is superior in Cost, Efficiency, Weight, Steering, and Space while the Fiat X-600 is superior in Brakes, Maintainability, Standardization, and Complexity. Overall, the CCS-I transmission rated higher and is recommended for the track-propelled LVTPX12. The summary of the ratings is shown in Figure 8-33. The calculations for the rated transmission as well as others for comparison are in Appendix D.

8.7.5 Auxillary-Propelled Vehicle Transmission Selection Criteria. The criteria used for this evaluation are the same as the previous analysis except for slight changes in Cost, Efficiency, Steering, and Ratio Coverage.

The difference in the Cost factor is due to the slightly lower cost of the transmissions due to less fuel required at 8 MPH.

Efficiency rates the amount of power required to turn the transmission at maximum engine speed when it is in neutral. No power required is ideal and scores 8 points; one point is deducted for each 5 HP required.

If the transmission has a torque converter lockup or equivalent for land mode, it scores 2 points.

There is no need for Steering function on water so the transmissions are rated for land performance.

	WEIGHTING FACTORS		FLAT X-500 RATING	FLAT X-500 SCORE	CCS-1 RATING	CCS-1 SCORE
1	COST	16	2.0	32	5.9	94.5
	MILLIONS OF DOLLARS		54.7		39.4	
2	EFFICIENCY	11	5.6	61.6	8.6	94.6
	POWER LOSS (HP)		62		47	
3	WEIGHT	11	7.0	77	9.5	104.5
	POUNDS		2685		1695	
4	STEERING	8	5.0	40	9.0	72
	LAND STEERING RATIO		1.2		1.4	
	WATER STEERING RATIO		1.6		2.2	
5	SPACE	8	5.5	44	9.8	78.4
	HEIGHT (INCHES)		18		15	
	VOLUME (CU.FT.)		25		12.5	
6	RATIO COVERAGE	6	10	60	10	60
7	RELIABILITY	6	1.0	6	1.0	6
8	BRAKES	5	9.9	49.5	7.8	39.0
9	AVAILABILITY	5	0	0	2.0	10.0
10	MAINTAINABILITY	5	5.5	27.5	4.5	22
11	STANDARDIZATION	5	1.5	7.5	0	0
12	COMPLEXITY	4	8	32	7	28
13	MULTI-SOURCE	4	8	32	8	32
14	HUMAN FACTORS	3	8	24	7.5	22.5
15	SENSITIVITY TO ENVIRONMENT	3	8.5	25.5	8.0	24.0
	TOTAL	100		519		688

Figure 8-33 Track-Propelled LVTPX12 Transmission Trade Off

- Fully modulated steer - scores 3 points.
- Land steering ratio in high of 1.4 scores 4 points; 1 point is deducted for each 0.2 unit difference.
- Pivot steer in neutral - scores 2 points.
- True pivot about vehicle center - scores 1 point.

<u>Ratio Coverage</u> is not needed on water, so only the land mode is rated.

- Top speed 36 MPH - scores 2.5 points.
- Speed on 60 percent slope of 2.5 MPH - scores 2.5 points.
- Speed in reverse of 2.5 MPH on the 60 percent slope and 8 MPH on the level - scores 2.5 points.
- Tractive effort at vehicle stall equals vehicle weight - scores 2.5 points.

All the factors are given the same weighting as in the Track-Propelled Vehicle Transmission Trade-off.

8.7.6 <u>Auxiliary-Propelled Vehicle Transmission Selection</u>. The three transmissions remaining from the initial selections are the Flat X-600, the CCS-2, and the HST-801A. Applying the rating scheme results in the CCS-2 scoring highest, the HST-801A next, and the Flat X-600 lowest. The CCS-2 was best in <u>Efficiency</u>, <u>Weight</u>, and <u>Steering</u>. The CCS-2 and the HST-801A were very close in <u>Space</u> and <u>Cost</u>, and the Flat X-600 was best in <u>Brakes</u>, <u>Maintainability</u> and <u>Standardization</u>. Based on the highest overall rating, the CCS-2 is recommended for the Auxiliary-Propelled LVTPX12. The summary of the ratings is shown in Figure 8-34. The calculations for the rated transmission as well as others are in Appendix D.

	WEIGHTING FACTORS		FLAT X-600 RATING	FLAT X-600 SCORE	CCS-2 RATING	CCS-2 SCORE	HST-801A RATING	HST-801A SCORE
1	COST	16	2.0	32	5.9	94.4	6.4	102.4
	MILLIONS OF DOLLARS		54.7		39.4		37.7	
2	EFFICIENCY	11	3.6	39.6	8.4	92.4	5.5	60.5
	POWER LOSS IN NEUTRAL		32		8		20	
3	WEIGHT	11	7.0	77.0	9.8	107.8	8.9	97.9
	POUNDS		2685		1595		1940	
4	STEERING	8	8.0	64.0	10.0	80.0	9.0	72.0
	LAND STEERING RATIO		1.2		1.4		1.32@ 36 MPH 1.52@ 24 MPH	
5	SPACE	8	5.5	44.0	9.8	76.4	9.3	74.4
	HEIGHT (INCHES) VOLUME (CU.FT.)		18.0 25.0		15.0 12.5		14.7 20.5	
6	RATIOS	6	10.0	60.0	10.0	60.0	10.0	60.0
7	RELIABILITY	6	1.0	6.0	1.0	6.0	1.0	6.0
8	BRAKES	5	9.9	49.5	7.8	39.0	8.8	44.0
9	AVAILABILITY	5	0	0	2.0	10.0	2.0	10.0
10	MAINTAINABILITY	5	5.5	27.5	4.5	22.5	3.5	16.5
11	STANDARDIZATION	5	1.5	7.5	0	0	0.2	1.0
12	COMPLEXITY	4	8.0	32.0	7.0	28.0	8.0	32.0
13	MULTI-SOURCE	4	8.0	32.0	8.0	32.0	6.0	24.0
14	HUMAN FACTORS	3	8.0	24.0	7.5	22.5	8.0	24.0
15	ENVIRONMENT	3	8.5	25.5	8.0	24.0	8.0	24.0
	TOTAL			521		695		649

Figure 8-34 Auxiliary-Propelled LVTPX12 Transmission Trade-Off

8.8 Track-Propelled Vehicle Design. The engine and transmission are selected and the preliminary designs and layouts of their necessary subsystems as included in the following paragraphs.

8.8.1 Engine. The General Motors Corporation Detroit Diesel Model 12V71T engine is installed just forward of the bulkhead between the troop and crew compartments. This is a two-stroke, turbo-charged, diesel engine with aluminum block. The power curve and fuel map are shown in Figure 8-29. The engine installation is shown in the General Arrangement Drawings in Section 5.0. With the remote transmission, a vibration damper is used to couple the propeller shaft to the engine. The engine mounts are bolted to the two longitudinal stringers, one on each side of the engine near the back of the engine and the third is in the front. Rubber elements are used as vibration isolators. It is planned that the mounts would be a slip-in fit at the front and have two bolts at the rear in order to simplify engine removal. The type of mount system now used on the M60 tank would serve as a model.

The engine controls consist of accelerator pedal, hand throttle, engine shut off and emergency engine shut off. The accelerator pedal and throttle are connected together so that either could be used. A flexible cable is used to connect them to the fuel control on the engine. The engine shut off is the normal means to stop the engine and it functions by stopping the fuel flow at the engine. This control uses a push-pull cable. The emergency shut off releases a flapper valve in the entrance to the air box on the engine and shuts off the air. This control provides a sure, second engine stop method and would be used if some malfunction in the injection system occurred or if

a fuel (or oil) leak sprayed fuel (or oil) into the air intake. The emergency shut off has to be reset on the engine which forces the operator to look at the engine. The control is actuated from the driver's station by a flexible cable. The description and function of the controls in the driver's station are in Section 18.0. The operation and maintenance are in Section 20.0.

The installation is shown in the General Arrangement Drawings in Section 5.0.

The engine crankcase is vented into the exhaust pipe after the turbocharger.

For all transmission control positions except the "Marine" position, the engine power is limited to 370 gross HP by an interconnection between the transmission control and the engine fuel control. The reasons for doing this are explained in Paragraph 8.8.2.

8.8.2 Transmission and Final Drives. The evolution and selection of the transmission has been covered in previous paragraphs. The new design transmission and final drive (CCS-1) are installed in the aft end of the vehicle under the floor. An automotive-type propeller shaft connects the engine and transmission. Some pertinent details of the transmission are presented below:

Chrysler Concept Transmission CCS-1 -

Ranges:	4 Forward		2 Reverse	
	1st	7.51:1	1st	7.55:1
	2nd	2.71:1	2nd	2.75:1
	3rd	1.77:1		
	4th	1.22:1		

Rating: Forward 1st, 3rd, and 4th and Reverse 1st

800 lb. ft., 2500 RPM, 340 HP

Forward and Reverse 2nd; torque converter operating

800 lb. ft., 2500 RPM, 340 HP

Forward and Reverse 2nd; torque convertar in lockup and in water mode

1750 lb. ft., 2500 RPM, 750 HP

Range Control:	Automatic and Manual
Steering:	Differential, Hydrostatically controlled
Converter:	13 inch diameter with Lockup
Dry Weight:	Including Final Drives; 1500 pounds

Figure 8-25 is a schematic of the transmission.

The transmission consists of a forward-reverse and range section, steering section, and final drives containing the combining gear sets as well as the final reduction. The output of the forward-reverse and range section is a shaft connected to the ring gears of both final drives. The torque converter is a standard, commercially available 13-inch, three-stage unit with lockup clutch. The stall torque ratio is 2.32 to 1. A special low-profile housing is used to minimize the height.

The forward-reverse and range section uses helical-constant mesh gears. Ratio changing is accomplished with two standard quick-acting (0.6 seconds) double hydraulic clutches. Reverse is provided a standard double clutch. There is a front oil pump to provide hydraulic pressure for clutch actuation and pressurizing the converter. The rear oil pump supplies lubrication oil and permits push starts to be made. The governor (a standard commercial unit) is driven from the same gear.

The hydrostatic steer has two hydraulic pumps and two hydraulic motors. Power from the engine is transmitted through the range section to the hydraulic pumps by means of a spiral bevel drive. The hydraulic steering motors are connected to the sun gear of both final drive planetaries. There is an idler incorporated in one of the final drives to reverse the direction of one sun gear. Rotation of the hydraulic motors cause one final drive to speed up and the other to slow down. If the transmission is in neutral, rotation of the motors will cause a pivot turn and, if the service brakes are locked, locking the ring gears in the final drives, the vehicle will make a perfect pivot turn about its center. This cannot be accomplished by the other transmissions discussed earlier. The combination of the hydraulic motors and the two final drive planetaries form a no-slip differential. With an increase in slippage of one track, the other will still transmit full power.

The hydraulic system has two variable displacement pumps, a supercharging pump, two fixed displacement motors, and necessary valves. The pumps are driven by the input to the torque converter and the speed of the pumps is always a function of engine speed. At any given engine speed and steering pump displacement, the radius of turn will increase with each step-up in transmission gear range. For any given steering pump displacement and transmission ratio, the turning radius remains constant regardless of engine speed because, as the engine speed decreases, the track speed decreases in direct proportion. Control of the variable displacement pumps is by conventional steering wheel through rods and linkage. The centering mechanism is at the transmission to insure that transmission movement does not cause the vehicle to steer.

The maximum steer ratios with the engine running at maximum speed are:

- 1st gear pivot steer available
- 2nd gear 2.2 to 1
- 3rd gear 1.64 to 1
- 4th gear 1.4 to 1

These ratios are the maximums and are infinitely variable to zero. With the transmission in neutral, the engine at full speed, and full steer, the vehicle will spin about its axis at the rate of one turn in approximately 3 seconds.

The air-cooled mechanical brakes are used as service and parking brakes. The linings are iron-sintered to steel backing plates which are welded to the shoes. Two disc brakes on a common shaft are used to get sufficient capacity in the limited space. The discs have an aluminim center section shaped to pump air through the two hardened-steel brake surfaces riveted to each center section. This type of brake has been successfully used on control differentials used to steer tracked vehicles. A hydraulic slave cylinder actuates the mechanical brake. To remove a tranmission, the slave cylinder is disconnected from the brake and the hydraulic system remains with the vehicle.

The transmission is mounted at three points; two trunnions at the rear and one swivel point at the front. This method of mounting has proven to be very desirable in that no external deformations can be transferred into the transmission housing.

The throttle valve control is connected to the engine fuel control by links and rods. Flexible cables have not been entirely satisfactory for this application in the past.

The quadrant at the driver's station has 6 positions: R-2 (reverse, 2nd gear), R-1 (reverse, 1st gear), N (neutral), L (low), D (drive), and M (marine, which is forward 2nd gear). The transmission control is interconnected with the engine fuel control so that the engine power is restricted to 370 gross HP under all conditions except when the transmission is in the "M" (marine) position and the Marine-Land switch is in the Marine position. In the "M" position, with the Marine-Land switch in the Marine position, the transmission is in 2nd gear forward, torque converter is locked up, and the engine fuel control allows full power. The 370 HP level provides good land performance (reference: Paragraph 8.10), and extends the life of all drive train components over what it would be if full power were available. This power setting can be adjusted up or down as needed during the design phase.

The transmission control quadrant in the driver's station is connected to the transmission by a low-friction, push-pull cable.

The transmission controls at the driver's station are described in Section 18.0, and the operation of the controls are described in Section 20.0. More detail on the transmission is in Appendix D. Figure 8-35 is a drawing of the range and steering portion of the transmission. Maintenance is discussed in Section 20.0. The final drives contain the combining planetaries which provide the steering function as well as speed reduction for the sprocket drive. The drives are completely self contained and mounted outside the hull. The two final drives are identical with the exception that one inner housing contains the idler in the steering system. Figure 8-36 is a drawing of the final drives. The General Arrangement Drawings in Section 5.0 show the installation of the transmission and final drives.

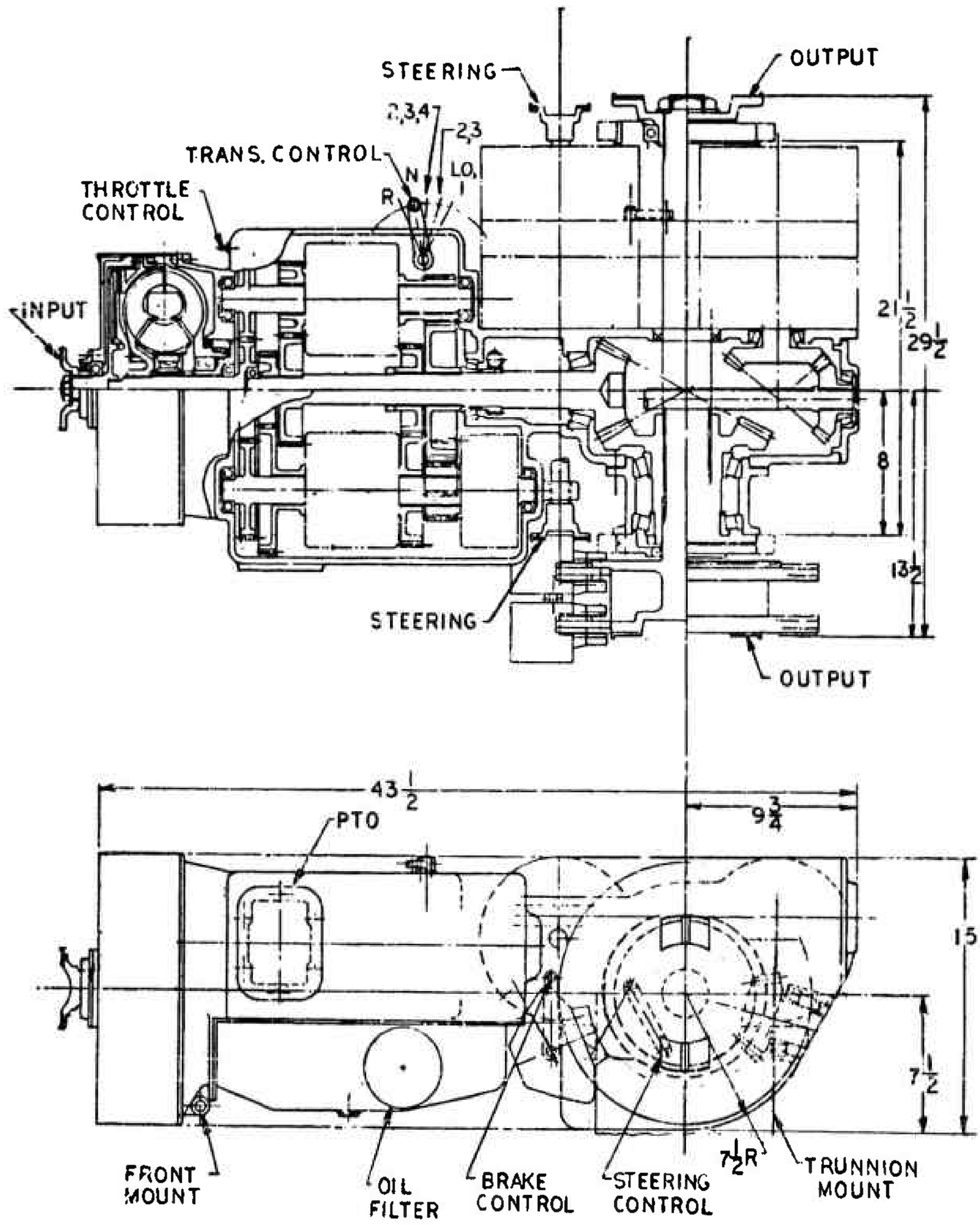


Figure 8-35 CCS Transmission Layout

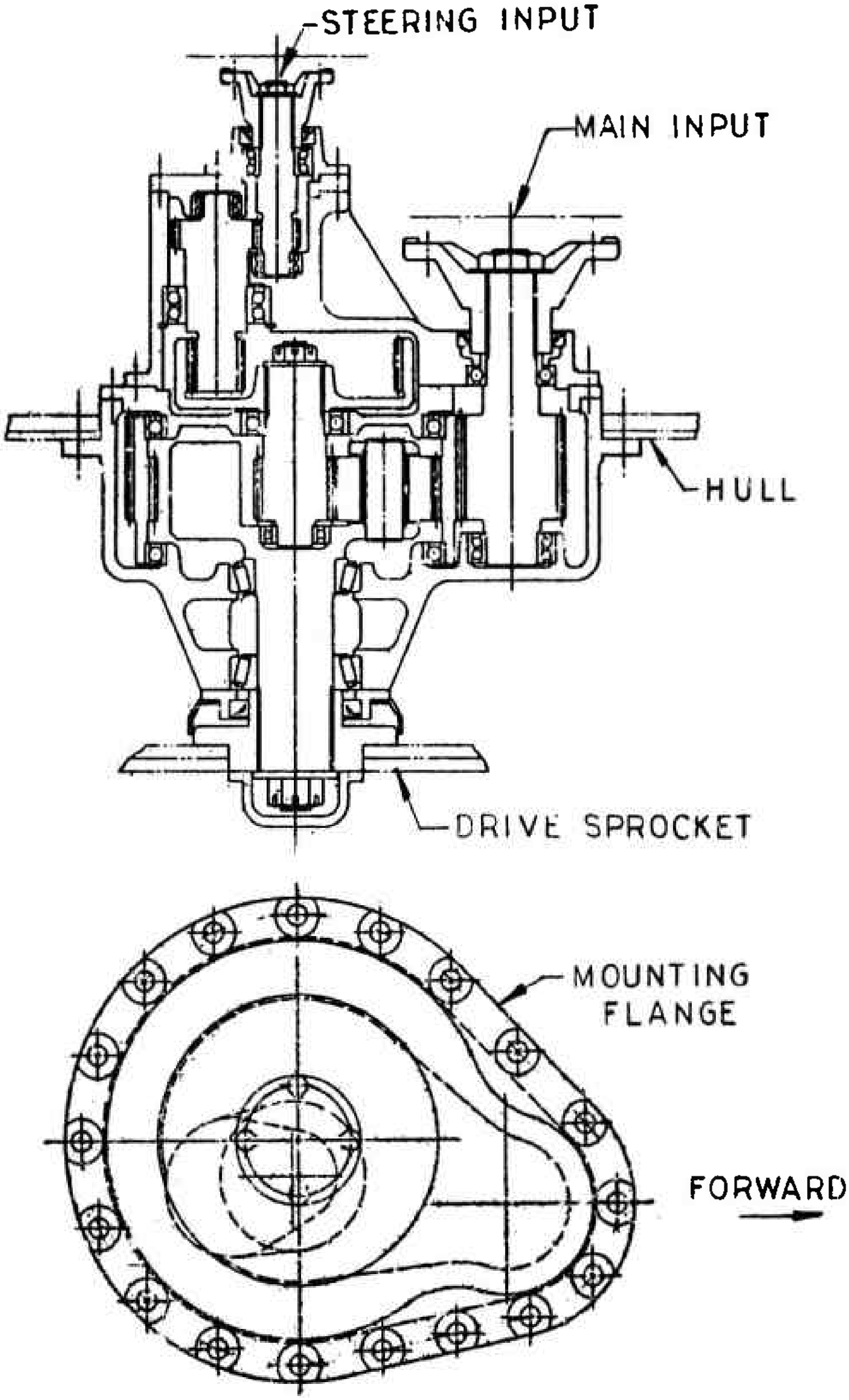


Figure 8-36 Fin ! Drive

8.8.3 Drive Line. The drive line is very simple and consists of five propeller shafts, one between the engine and transmission and two on each side of the transmission that go to the final drives. The engine to transmission shaft is a two-piece shaft with center bearing and sliding spline to ease assembly. The tubes are 3 inches diameter and 0.083 inches wall thickness. Conventional universal joints are used on each end of the shaft and at the center. The sizes recommended are shown in Figure 8-37. The joints and splines are readily available and can be multi-sourced. The installation is shown in the General Arrangement Drawings in Section 5.0.

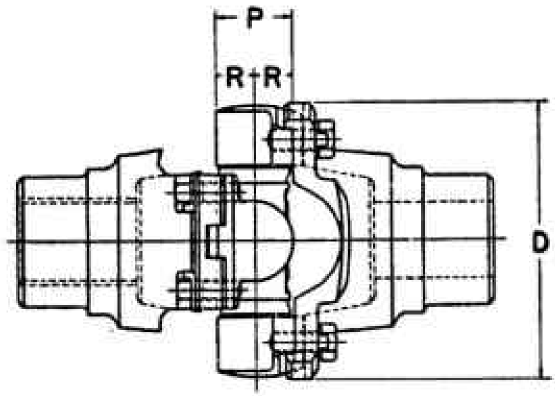
8.8.4 Engine and Transmission Cooling. (See Section 15.0).

8.8.5 Aspiration and Exhaust Systems. (See Section 14.0).

8.8.6 Lubrication System. There are only two lubrication systems in the vehicle; the engine and the transmission. Each system is completely self contained. Since the final drives are functionally part of the transmission, the two units share a common lubrication system. The transmission lubrication system consists of the sump, scavenge and charge pumps, connecting lines, and the oil cooler on the engine. Engine oil, SAE 10W, is used in the transmission for temperatures above -25°F and sub-zero oil for temperatures below -25°F.

8.8.7 Fuel System. (See Section 13.0).

8.9 Auxiliary-Propelled Vehicle Design. Most of the machinery installations in the auxiliary-propelled vehicle are the same as the track-propelled vehicle. The exceptions are described in the text following.



	TRANSMISSION STEERING SHAFT	ENGINE TO TRANSMISSION SHAFT	TRANSMISSION OUTPUT SHAFT
SWING DIAMETER D	$4\frac{1}{2}$"	$5\frac{27}{32}$"	$8\frac{5}{8}$"
BEARING HEIGHT R	$\frac{39}{64}$"	$\frac{11}{16}$"	1"
FACE TO FACE OF YOKES P	$1\frac{7}{32}$"	$1\frac{3}{8}$"	2"
STATIC TORQUE CAPACITY	2100 FT. LBS.	4000 FT. LBS.	14000 FT. LBS.
APPROXIMATE WEIGHT	6#	11.5#	35#

Figure 8-37 Universal Joints

8.9.1 Engine. This is the same as the track-propelled vehicle except for the transfer case on the aft end for auxiliary-propulsion drive and a difference in the engine power limiting control. The engine power is limited to 370 gross HP at all times except when the transmission control is in neutral and the Water-Land switch on the instrument panel is in the Water position. The installation is shown in the General Arrangement Drawings in Section 5.0. The driver's controls and operation are described in Sections 18.0 and 20.0, respectively. Maintenance is in Section 20.0.

8.9.2 Transmission and Final Drives. These are the same as used in the track-propelled vehicle with the exception that the transmission is slightly smaller and lighter (1400 pounds dry) and the controls are a little different. The shift quadrant at the driver's station has the positions: R-1 (reverse, 1st gear), R-2 (reverse, 2nd gear), N (neutral), F-1 (forward, 1st gear), F-2 (forward, 2nd gear), and D (drive, transmission will shift up to 4th gear). When in the water mode, even though the transmission is in neutral, the vehicle can be steered. If the steering wheel is turned, the tracks on one side will move forward and the other side will move backwards. This could be used to supplement the propeller steering system if desired by the driver. The driver's controls are described in Section 18.0. The operation and maintenance are discussed in Section 20.0. The installation is shown in the General Arrangement Drawings in Section 5.0.

8.9.3 Drive Line. The propeller shafts between the engine and transmission and between the transmission and final drives are the same as used in the track-propelled vehicle. The installation is shown in the General Arrangement Drawings in Section 5.0. The auxiliary propulsion unit and drive line are unique to this vehicle.

8.9.3.1 Outboard Propellers. Two controllable pitch propellers, one on each side at the stern, furnish the propulsion means in the water mode. The pitch of the propellers can be varied individually from full ahead to full astern at the driver's option. This supplies the steering and the reverse function. With the engine running at full speed, the driver controls the direction and speed with the propeller pitch levers and has the option of full thrust or no thrust. The engine governor regulates the fuel flow according to the power demanded by the propellers. The three-bladed propeller of 23.5 inches diameter is in a shroud ring that has a maximum diameter of 25.5 inches. The unit is swung on the drive shaft axis from the stowed position to operating position by a hydraulic actuator. (See Section 12.0, Paragraph 12.4.5 for details on the propeller actuation system. Figure 8-38 is a sketch showing the location of the propeller and the way it is moved into operating position.

Two sets of bevel gears (1:1 ratio) transmit the power from the drive shaft in the vehicle through the strut and to the propeller. Figure 8-39 is a sketch of the propeller drive unit.

The blade pitch is controlled by a hydraulic actuator in the stationary hub which moves the control rod connected to the blade links. The controls at the driver's station are linked by a Controllex cable to the hydraulic valve at the propeller drive. The discussion of the driver's controls is in Section 18.0.

The operation and maintenance are in Section 20.0. The performance analysis and trade-off analysis are in Section 4.0.

The installation is shown in the General Arrangement Drawings in Section 5.0.

8.9.3.2 Vehicle Drive Train to the Propeller Unit. The sketch of the drive line in Figure 8-40 illustrates the approach that is taken. The power is split at the transfer case with a shaft running down each side of the vehicle under and behind the troop seats.

The transfer case contains the hydraulic disconnect clutch that disengages the propeller drive. A train of helical gears with a bevel set on top accomplish the changes of direction and location. Recent developments in chain drives make this form of power transfer a possibility in the future. At present, it was decided that gears would be the most reliable for the space available.

The angle drive boxes are simple bevel gears.

The connecting propeller shafts are conventional automotive-type, very similar to those recommended for the transmission drive line. The short shafts forward connecting the transfer case and the angle drives are one-piece shafts with sliding splines and use the same universal joints as the engine-to-transmission shaft. The shafts between the angle drives and the outboard propeller drives are two-piece shafts with center bearing, three universal joints of the same size as the engine transmission shaft, and with a sliding spline each.

The operation and maintenance are in Section 20.0. The driver's controls are discussed in Section 18.0. The installation is shown in the General Arrangement Drawings in Section 5.0.

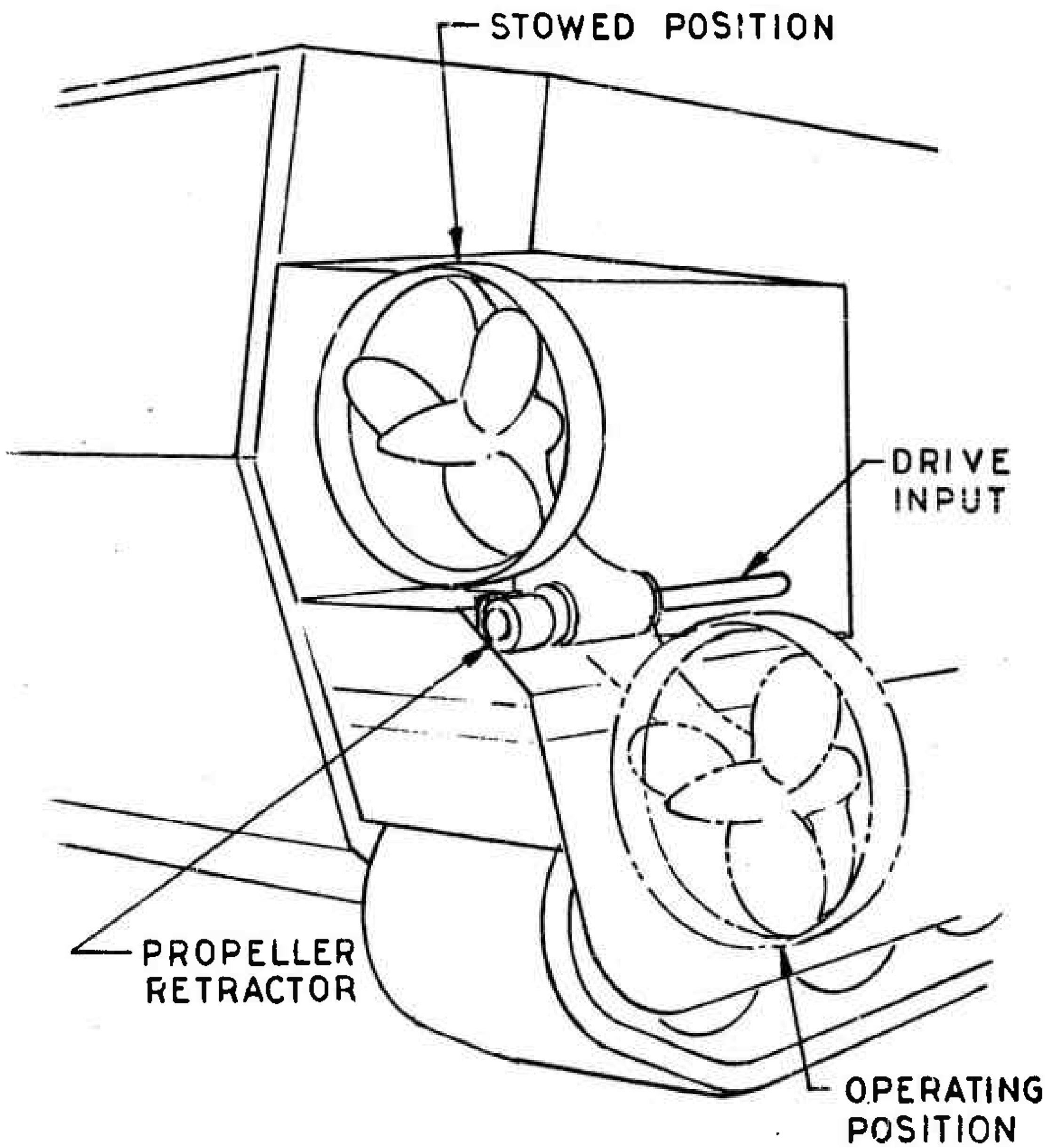


Figure 8-38 Propeller Unit

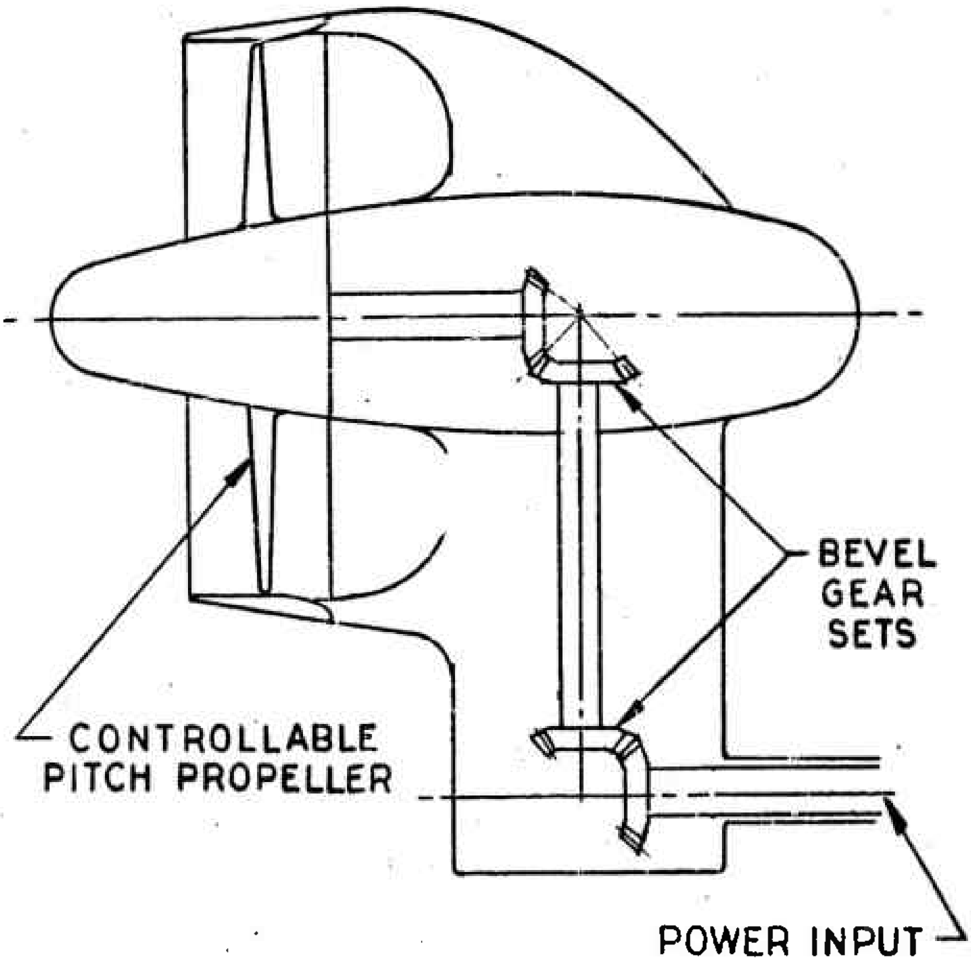


Figure 8-39 Propeller Drive Unit

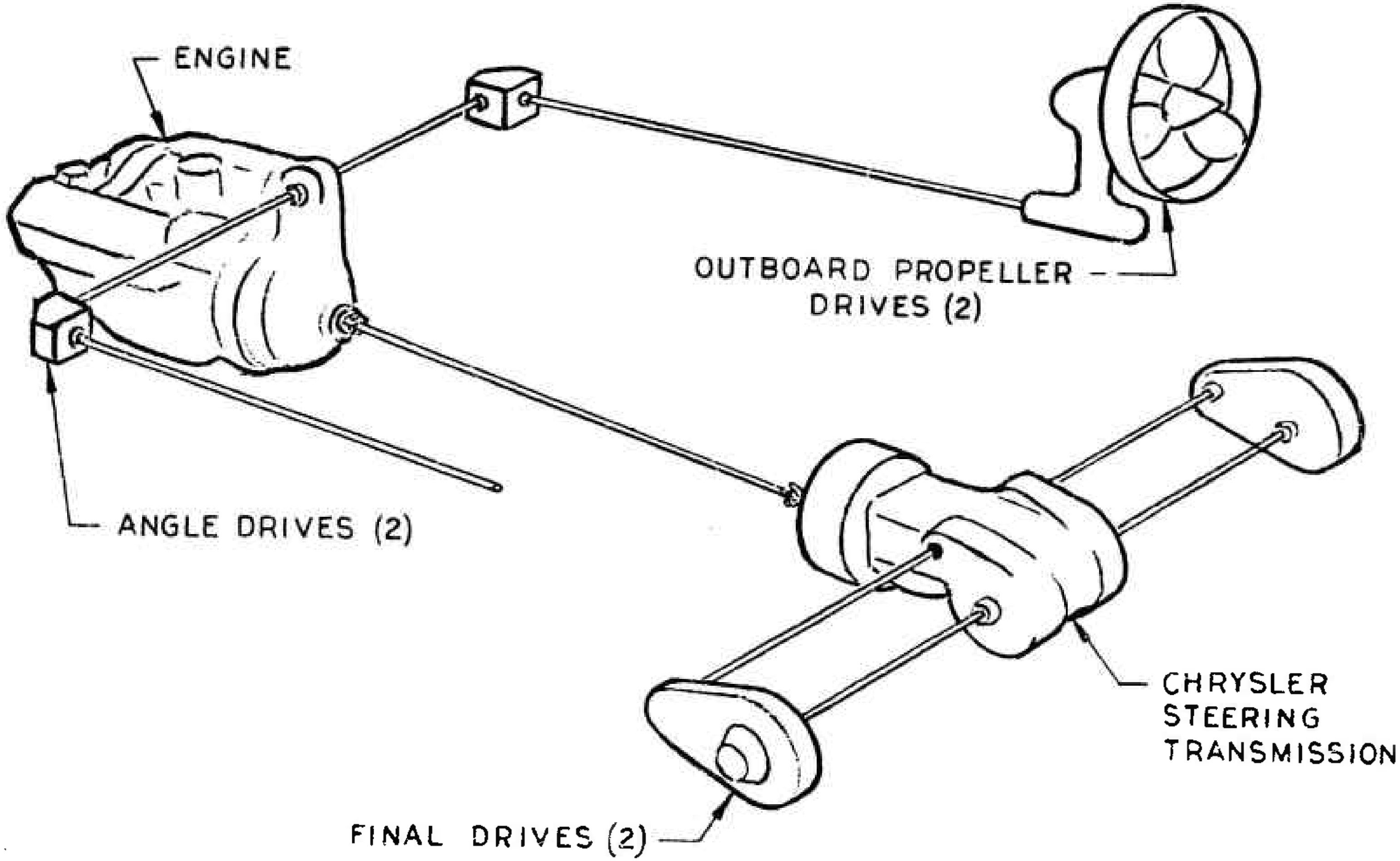


Figure 8-40 Power Train Schematic - Auxiliary Propulsion

8.9.4 Cooling System. This is the same as the system for the track-propelled vehicle and is described in Section 15.0.

8.9.5 Aspiration and Exhaust System. This is the same as the system for the track-propelled vehicle and is described in Section 14.0.

8.9.6 Lubrication System. The lubrication system for this vehicle is the same as for the track-propelled vehicle except that the propeller drive system is added. Each unit of the propeller drive has its own separate lubricating system and there is no tie with any other unit.

8.9.7 Fuel System. (See Section 13.0).

8.10 Vehicle Performance. The land and water performance values for the 2 versions of the LVTPX12, along with LVTP5 values, are listed in Figure 8-41. Both versions of the LVTPX12 exceed the specified performance, and surpass the LVTP5 performance by a sizeable amount. The land performance curves for the auxiliary-propelled version are on Figure 8-42. The only difference between the two versions is the vehicle weight and the heaviest vehicle is shown.

	SPECIFICATION MINIMUM	SPECIFICATION DESIRED	LVTP5	LVTPX12 TRACKED	LVTPX12 AUXILIARY
LAND					
GROSS VEHICLE WEIGHT (LB)			81,780 e	51,990	53,670
SPEED LEVEL, ROAD (MPH)	30		28.5 b 30 e	36	36
SPEED, 10% SLOPE (MPH)			10.5 c	15.5	14.5
SPEED, 60% SLOPE (MPH)	2.0		2.1 c	2.5	2.5
TRACTIVE EFFORT AT STALL (LB)			77,000 c	52,000	53,700
RATIO, TRACTIVE EFFORT / VEHICLE WEIGHT			0.94 c	1.00	1.00
SPEED REVERSE LEVEL ROAD (MPH)	8.0		6.5 b	16.0	16.0
SPEED REVERSE 60% SLOPE (MPH)	2.0		2.2 c	2.5	2.5
MINIMUM SUSTAINED SPEED LEVEL, ROAD (MPH)	5.0		3.0	2.0	2.0
TIME TO ACCELERATE TO 20 MPH ON LEVEL ROAD (SECONDS)				12	12
WATER					
SPEED, FORWARD, CALM WATER (MPH)	8.0	10.0	6.7 d 6.8 e	8.1	10.7
SPEED, REVERSE, CALM WATER (MPH)	3.5		2.6 d	5+	5+

e Characteristics Sheet LVTP5A1 dated 23 Jan. 1964.

b "Report on Land Performance Tests - LVT-151", Ingersoll Products Div., Borg-Warner Corp., 11 Feb. 1953, Contract NObs-2716.

c "Report on Study, Performance and Fuel Consumption for the LVTP5", Ingersoll Kalamazoo Div., Borg-Warner Corp., Contract NObs-3597, 20 July 1955.

d EES Report 2J111X1603, 3 Nov. 1953.

\+ Model tests were not conducted at speeds above 5 MPH; reference Section 4.0.

Figure 8-41 Comparison of LVTPX12 and LVTP5 of Land and Water Performance

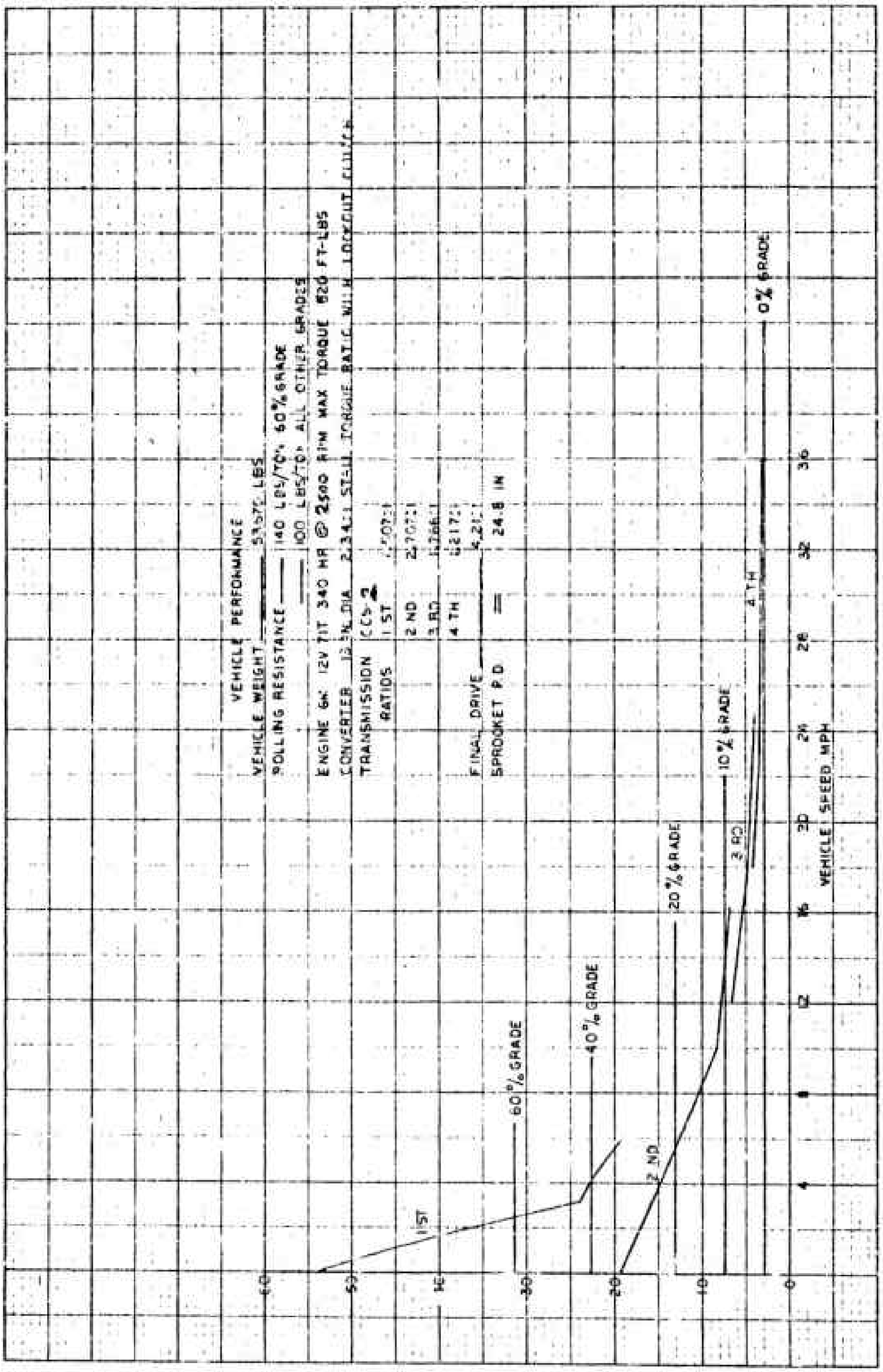


Figure 8-42 Auxiliary Propelled LVTPX12 Land Performance Curve